Power and The Engineer

Vol. 32 NEW YORK, AUGUST 16, 1910 No. 33

AMBITION to begin with and a little encouragement from others to brighten the flame has been the secret of many a man's success.

It is possible to kill ambition by withholding encouragement. The latter costs nothing, and frequently proves to be a good investment.

However, if a man has no desire to become greater, to perform better work, or to hold a more responsible position, he is not very likely to receive much encouragement, and may consider himself fortunate if he can hang on to the same old job.

In a Southern cotton mill power plant a young man, possibly eighteen or twenty years of age, holds a position as oiler. He works from five a.m. until six p.m. each work day for a wage of $8 per week.

He is not satisfied with his work, hours or wages, but, although laboring under conditions which would be obnoxious to most men, he lacks the ambition to prepare for something better.

The South holds out opportunities for engineers, but this oiler lacks the ambition to fit himself for any of them, and if someone were to force this man into a good position, he could not hold it.

It is doubtful if encouragement, and plenty of it, would have much effect in his case, because he lacks the incentive to get started in an upward direction.

He will see others come and go, prosper and obtain positions of trust and responsibility, while he is left with his oil can and greasy waste.

The ambitious man is bound to receive more or less encouragement in his work.. The last paragraph of the article entitled, "Economical Fire Room Methods," which appeared in the August 2 issue, illustrates this point.

Boys working as helpers and firemen have been taught to read and write, and were thus enabled to fill better positions.

The head fireman, because of his ambition and perseverance, has been promoted by his chief to the position of chief engineer of an up-to-date turbine plant.

Not because he was a favorite, but because he had ambition to become something better than a head fireman, which was undoubtedly stimulated by the encouragement he received from the chief.

The man without ambition is to be pitied.

The man with a burning desire to improve his position is likely to succeed.

And the man who "hands" out an encouraging word here and there exerts a silent but powerful influence over the other two.

Do not be satisfied with the position you hold while there is something better in view, and do not forget to encourage the little fellow who is struggling to gain a higher level. There is plenty of room for everybody.

Large Noncondensing Power Plant

By Warren O. Rogers

The power plant of the Birmingham Railroad, Light and Power Company is probably the largest central station in the United States running both reciprocating engines and turbines noncondensing. The exhaust steam is sold for heating purposes during the winter months. The power plant contains 31 water-tube boilers, totaling 17,200 horsepower; six Corliss engines, both double and compound, aggregating 9850 horsepower, and two 3000-kilowatt Curtis turbines. Both alternating and direct currents are generated.

Birmingham, Ala., is situated about midway between the Cahawba and Warrior rivers, each some six or seven miles distant. As a result, most of the steam plants in that city are run noncondensing, as condensing water is procurable only at a cost that practically prohibits its use without the installation of cooling towers or cooling sprays. It is for this reason that the steam-generating units of the Birmingham Railway, Light and Power Company's plant are run noncondensing.

This power plant furnishes current for the street-railway system and for commercial illuminating and power purposes throughout the city, as very few of the large office buildings are equipped with isolated plants. The plant has been remodeled, the transformation being from the old type of belt-driven unit to the modern direct-connected unit.

Engine Room

The engine room contains six Corliss engines, one of 500-horsepower, two of 1300-horsepower and three of 2250-horsepower capacity. The small engine has cylinders 20x30 inches and was originally used as a pair, but at present one side is disconnected and the two direct-current generators mounted on the engine shaft are driven by one engine. The generators are each of 2000 kilowatts capacity, at 125 volts, the armature revolving at a speed of 150 revolutions per minute. This unit carries the engine and boiler rooms lighting load, in case all the other generators are stopped, or it can be used as an exciter, when the motor-driven exciters are out of commission.

The two original 1300-horsepower engines now have 20 and 20 by 48-inch and 28 and 44 by 48-inch cylinders, one being a double and the other a cross-compound engine. Each of these engines is direct connected to an 850-kilowatt direct-current railway generator. The speed is 80 revolutions per minute. Of the three 2250-horsepower units, one is a double Corliss engine, with cylinders 36x60 inches. The other two are cross-compound Corliss engines, both having cylinders 36 and 60 by 60 inches. One of the cross-compound engines is direct connected to a 1600-kilowatt direct-current railway generator, while the other two units drive 1500-kilowatt alternating-current generators.

Besides the engine units there are two Curtis turbines driving 3000-kilowatt 60-

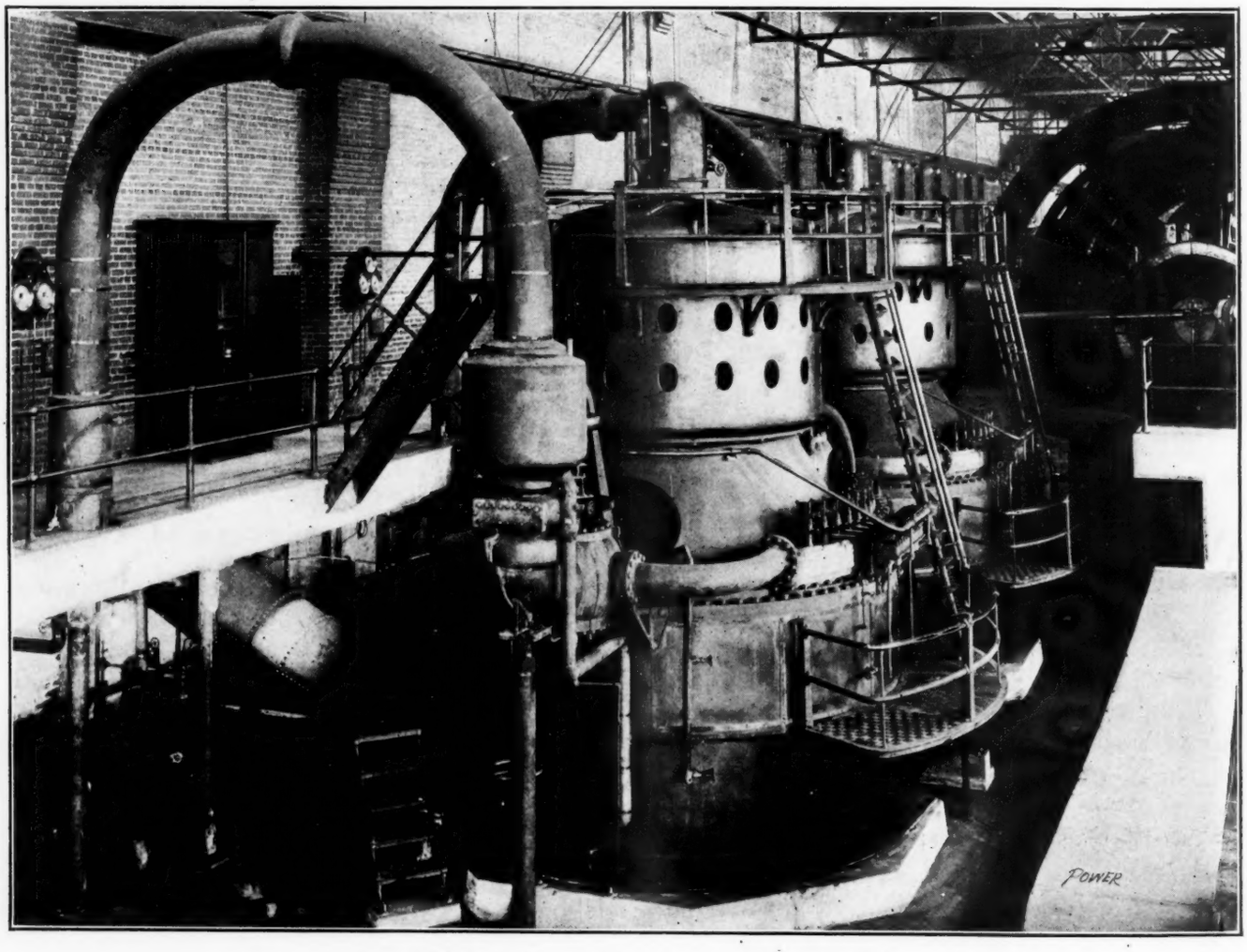


Fig. 1. View of Engine Room from the Turbine End

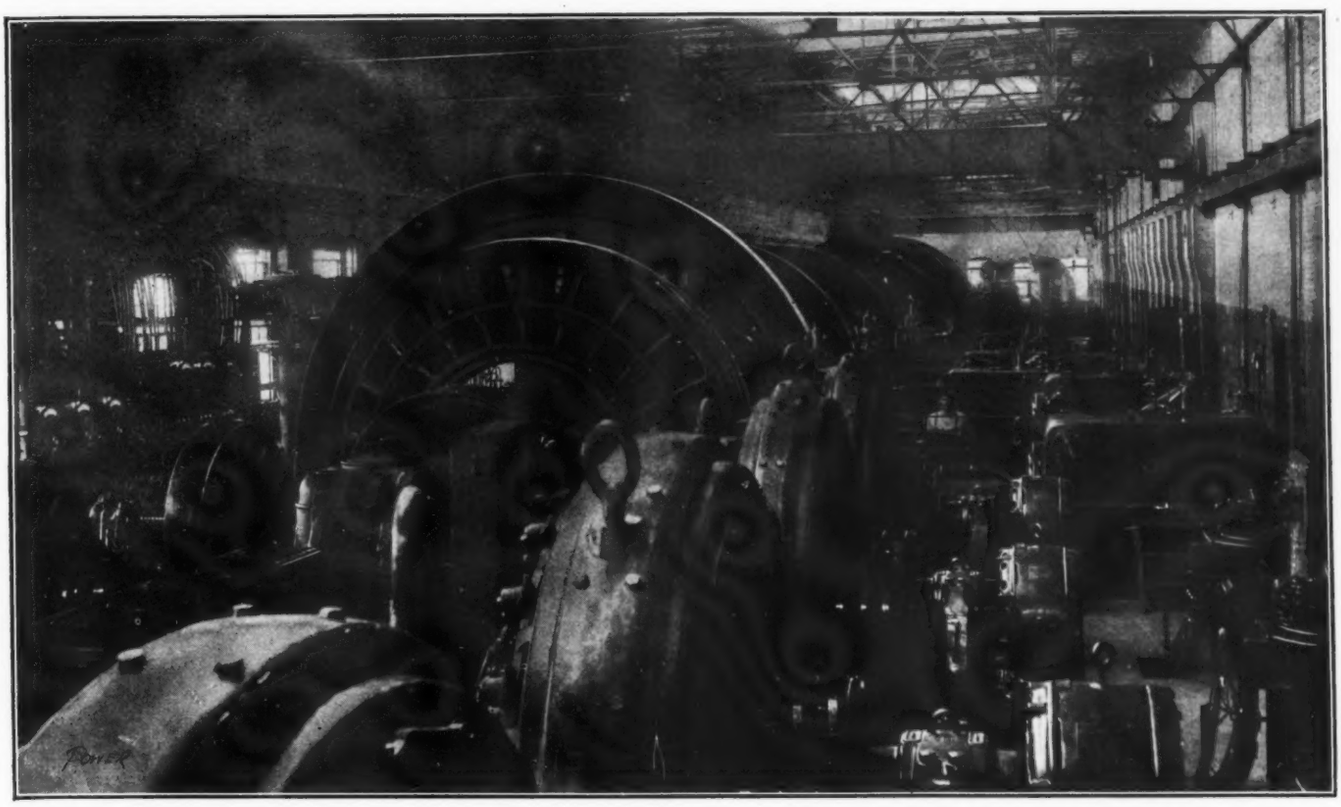


FIG. 2. VIEW OF ENGINE ROOM, SHOWING RECIPROCATING UNITS AND ROTARY CONVERTERS

cycle generators. All generators in the plant are of General Electric make. The controlling valves on the turbines are electrically operated. Fig. 1 shows the turbine units in the foreground and the reciprocating engines in the rear.

There are seven rotary converters, five being of Westinghouse make. Three of these are rated at 750 kilowatts and the other two at 1000 kilowatts.

The remaining two are General Electric machines. Two motor-driven exciter units furnish current for excitation, the smaller unit consisting of a 75-horsepower induction motor direct connected to a 125-volt generator. The larger exciter consists of a 175-horsepower induction motor driving a direct-current generator at 600 revolutions per minute.

In the pit below the engine-room flooring, and on a level with the floor of the turbine pit, is a 300-kilowatt railway booster, and just above it, on the main floor, are three 750-kilowatt transformers. At the extreme end of the plant are located nine 50-kilowatt and five 100-kilowatt transformers.

BOILER ROOM

In the boiler room, Figs. 4 and 5, are twenty-four 600-horsepower water-tube boilers, and seven 400-horsepower water-tube boilers all of Babcock & Wilcox make. They are arranged as shown in Fig. 3, and are all equipped with chain grates, with the exception of four 600-horsepower boilers. Of the total number of boilers twenty-one are fitted with the Dutch-oven furnaces, the remainder with furnaces of regulation design. The pressure carried on all of the boilers is 175 pounds per square inch.

There are three stacks, each 175 feet high, one being 10 feet in diameter and two 12 feet in diameter. The two larger

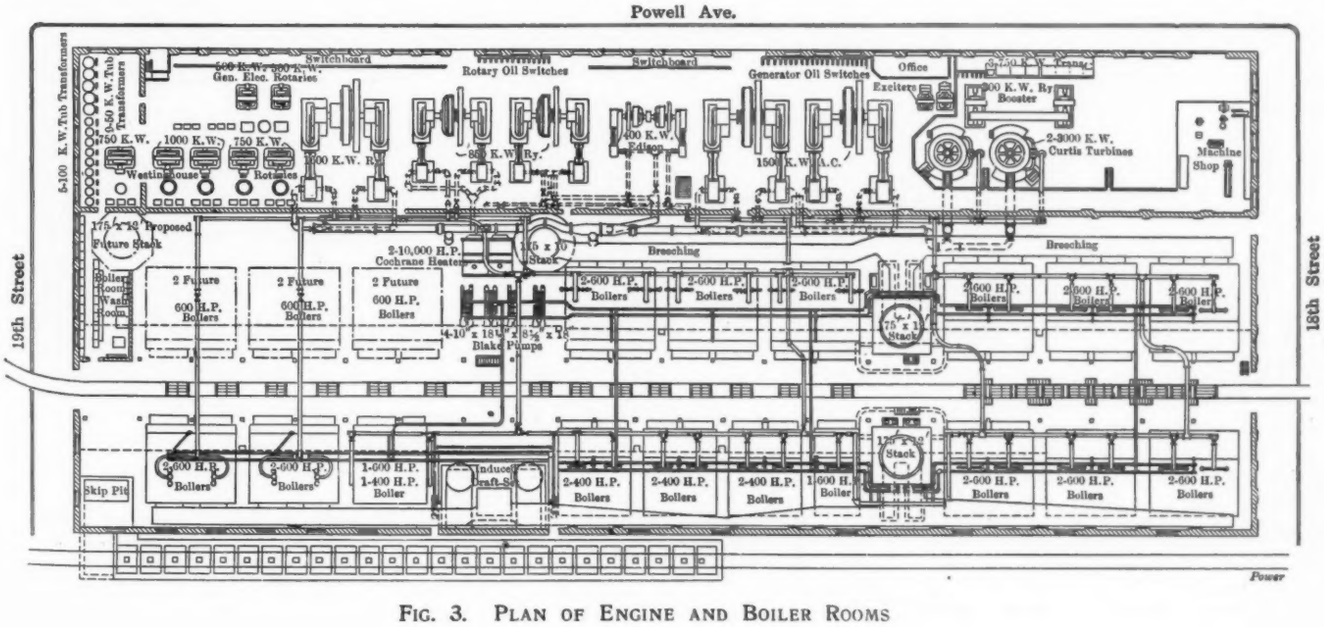


FIG. 3. PLAN OF ENGINE AND BOILER ROOMS

stacks are constructed of brick and the smaller of steel.

All stokers are driven from shafting running in front, and at the top of each row of boilers, each stoker being operated by an eccentric and rod, the latter being connected to the ratchet of the feeding device. These shafts can be driven by three 15-horsepower vertical Waches steam engines or four General Electric 7½-horsepower induction motors.

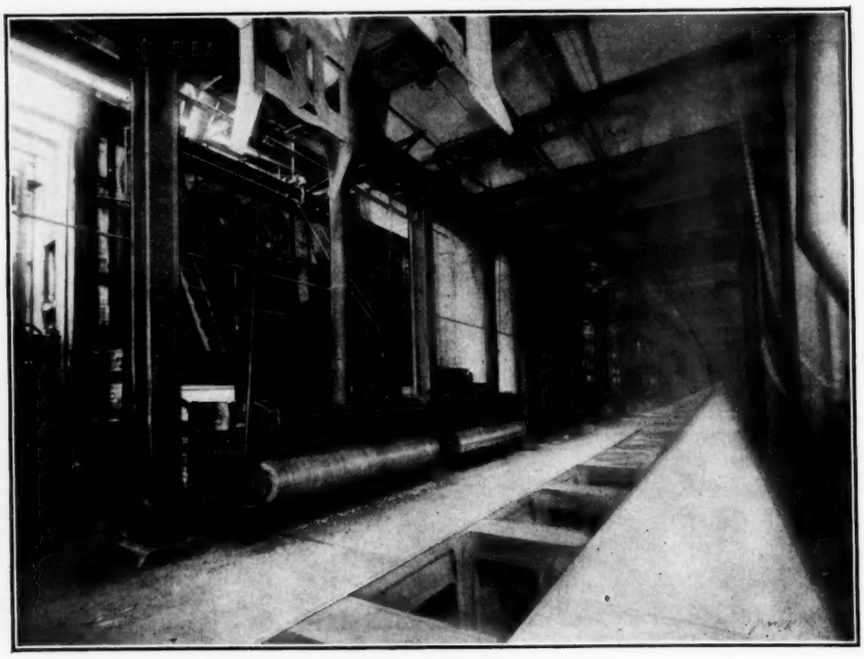


FIG. 4. PARTIAL VIEW OF BOILER ROOM

PIPING

A general plan of the piping arrangement for line and exhaust steam is presented in Fig. 3. The exhaust pipes from each turbine pass through the walls forming the partition between the boiler and engine rooms, and extend along the opposite side of the wall to the extreme end of the building. The engines exhaust into a common header which supplies steam for commercial heating and also for heating the boiler feed water in two 10,000-horsepower Cochrane heaters.

Exhaust steam is utilized for heating various buildings in the city during the winter months, so that operating the double engines and even the compound engines and steam turbines noncondensing has some advantages, and in this particular plant reduces the cost of the electrical energy per kilowatt-hour to a figure that compares favorably with some of the more modern and better equipped power plants.

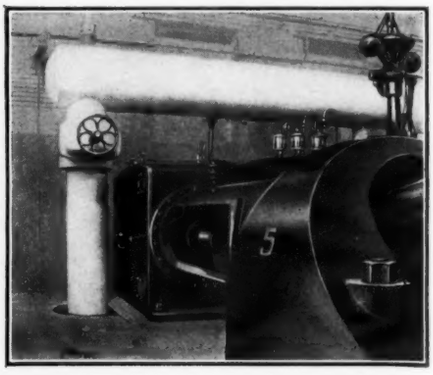


FIG. 6. POSITION OF RECEIVER

In the case of the double engine the exhaust steam passes to the heating main with a back pressure of 10 pounds per square inch. The cross-compound engines have the receiver located above the cylinders, as shown in Fig. 6. The steam for the heating system is not taken from the exhaust of the low-pressure cylinders but from a tee on the high-pressure exhaust pipe connecting the cylinder and the receiver.

Fig. 7 shows the arrangement of the boiler-feed pumps which are placed at the end of the shorter row of boilers, as shown in Fig. 3. They consist of four Blake pumps having 10x18½x8x18-inch cylinders, the water end being of the outside-packed type with pot valves. The arrangement of the steam and water piping is shown to good advantage.

The method of handling the coal and ashes will be described in a separate article.

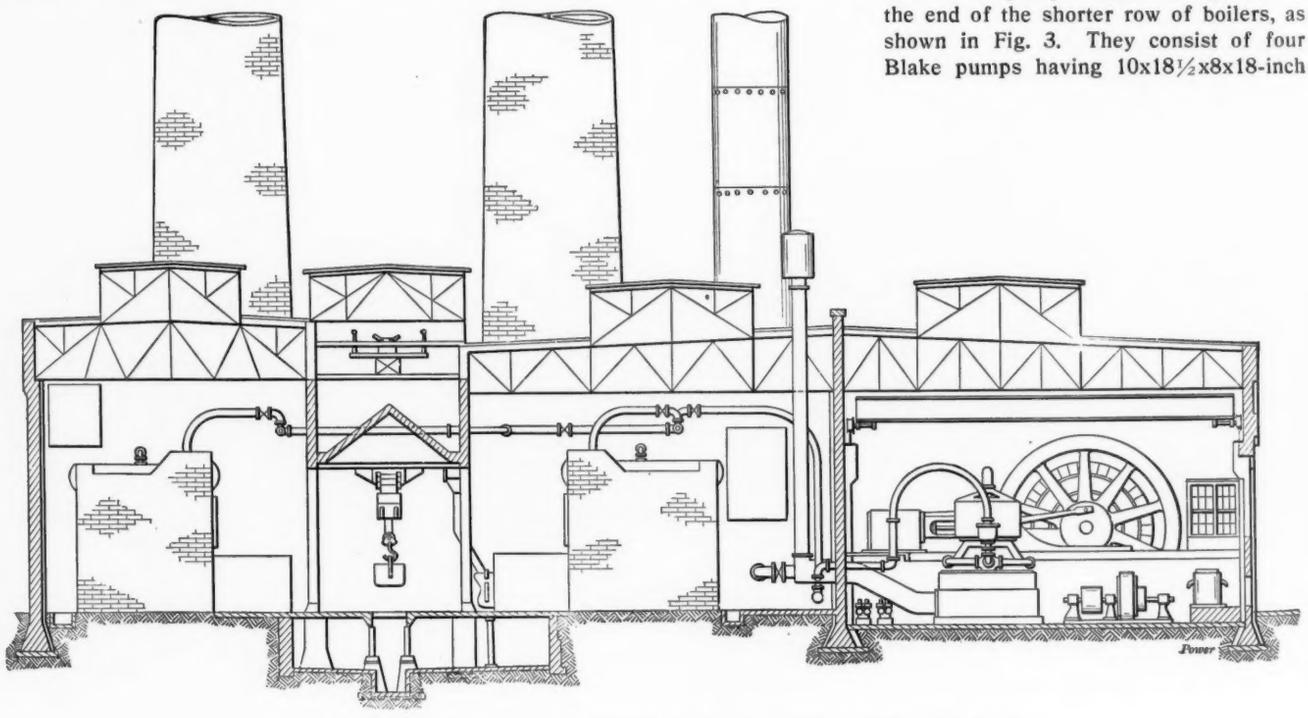


FIG. 5. SECTION THROUGH TURBINE END OF BOILER AND ENGINE ROOM

Probable Future Boiler Practice

By James E. Steely

A discussion of the heat distribution in a boiler and a plea for such design as will enable each part of the heating surface to contribute an equal share of the total evaporation.

The capacity of a boiler is a variable factor. Builders usually allow about 3½ pounds equivalent evaporation per square foot of heating surface in rating a boiler. Under the present standard of construction this is about the evaporation which is possible without forcing. However, no two sections of the boiler will do the same work; the part close to the fire will generate a large portion of the steam while that near the flue will generate very little. Some idea of the relative amounts done by the various rows of tubes was given by an instrument, used some time ago, to indicate circulation in the boilers of the Geological Survey plant at St. Louis. The bottom row of tubes were entirely inclosed in a fire-clay baffle; notwithstanding this protection from the heat, the machine made three times as many revolutions when placed in one of the bottom row of tubes as when placed in the tube immediately above it. When placed in the third row from the top it revolved very slowly. This demonstrates that the first row of tubes does many times as much work as any of the others and that the upper rows do very little. Now if the average evaporation is 3½ pounds of water per square foot of heating surface, the lower row may be evaporating 8 or 10 pounds, while the upper row is evaporating possibly only one-tenth of a pound.

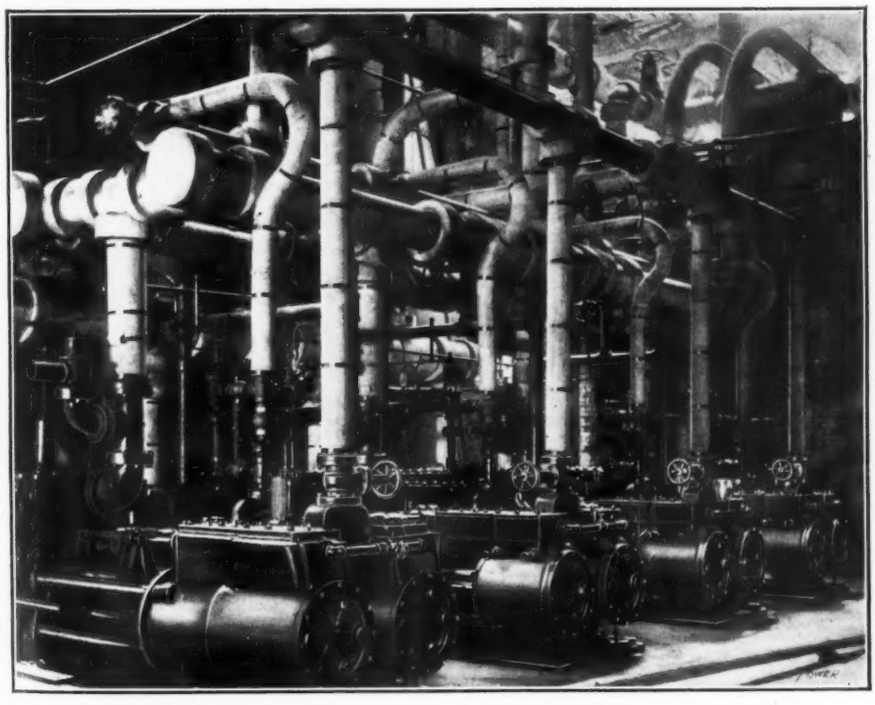


FIG 7. ARRANGEMENT OF FEED PUMPS AND PIPING

The size of the grate of the modern boiler is such that, when burning coal as rapidly as good practice and economy permit, only about 3½ to 5 pounds of water can be evaporated per square foot of heating surface. Some time ago, one of the New York power stations was equipped with a Babcock & Wilcox boiler having two Roney stokers, one placed under each end of the boiler, thereby doubling the grate surface. About 75 per cent. overrating was obtained by this arrangement without any considerable decrease in efficiency. This would indicate that the trouble with out modern standards is that the ratio of heating surface to grate surface is too large. The arguments for this practice of high evaporation are low first cost, economy of room and smaller loss due to radiation; those against it are increased maintenance cost and rapid deterioration of the boiler.

The capacity of a boiler is approximately proportional to the velocity of the gases and water. Thus the more rapid the circulation of water in the boiler or the more rapid the passage of the gases, the more rapid will be the generation of steam. Probably every engineer has seen diagrams showing the difference in ebullition between where there is circulation and where there is none. Theoretically the correct way to get the highest economy is to have the water circulating in the opposite direction to the hot gases, but the writer knows of no way in which this opposed current could be obtained and still allow free circulation of water in the boiler. If the circulation were restricted, the steam would be liberated with violence. It may be stated, however, that when a water-tube boiler of the Babcock & Wilcox or Stirling types is set with vertical baffles, this opposed current arrangement is secured to some extent. This is about as much as can be expected without sacrificing the free circulation of the water in the boiler, which is more important. The opposed flow should be secured by stages just as steam is expanded in doing work. It is out of the question to economically obtain a complete expansion of steam in one long cylinder, but the same end is arrived at with excellent results in two, three, or four stages. While there is no analogy between steam expansion and heat absorption, the same principle may be applied to obtain the best results. In other words, the boiler and economizer are better when separated than combined in one. If a two- or three-stage economizer were used, it would cool the flue gases as low as any boiler which could be built and with much less complication. Plants using noncondensing engines usually have enough exhaust steam to heat the feed water as hot as it is possible to handle it in the pump and when this hot water is further passed through a good economizer it is raised to almost the temperature of evaporation, leaving merely the work of evaporation to the boiler. If the exhaust steam were not available, a second economizer could be used in tandem, but the temperature of the flue gases would be so low as to seriously affect the chimney draft.

It has been stated that efficiencies as high as 90 per cent. are attainable. If one closely examines the heat balance of an ordinary boiler test, it will be seen that it is almost impossible to obtain any such result except perhaps with a very bulky and expensive layout. The "unaccounted for" losses and the loss due to incomplete combustion usually amount to over 10 per cent. and it would be necessary to return the burnt gases to the air at the same temperature as they were received. When temperatures are taken for calculating the chimney losses, they are taken before the gases pass through an economizer, and when the loss up the stack figures 18 to 25 per cent., it must be kept in mind

that the economizer will take up considerable of this apparently lost heat. Thus while the efficiency of the boiler might be only 65 per cent., the efficiency of the boiler and economizer might be 75 per cent. The results of recent investigations show that a high furnace temperature means a high stack temperature and that a large increase in the combustion-chamber temperature is accompanied by very little increase in efficiency.

The problem of heat absorption is discussed fully in the Geological Survey bulletin No. 325 and considerable experimental data as well as conclusions drawn therefrom are set forth. Everyone interested in heat absorption should study these experiments carefully as many of them result contrary to the logical method of reasoning.

In the writer's opinion a boiler should be proportioned so that all the heating surfaces would do equal work. If this result could be attained, two or three times as much steam would be generated per unit as that which is now possible.

Corliss Engine with Two Eccentrics

By Prof. R. C. Heck

A discussion of the advantages of operating the steam and exhaust valves by separate eccentrics, by which method a greater range of cutoff is obtained.

The purpose in using two eccentrics on a Corliss engine is to obtain a greater range of cutoff under the control of the governor. With one eccentric, cutoff by release of the valve in the normal manner cannot occur much later than at about three-eighths of the stroke. With a separate eccentric for the steam valves, the governor may control cutoff to as late as three-quarters of the stroke.

In Fig. 1 is represented a typical releasing gear, with some nonessential details omitted. Piece A is the oscillating arm or double crank, continuously and positively driven by the wristplate. It carries the hook claw B, with latch arm L and trip arm T. Pin P is on the valve arm D, which is keyed fast to the valve spindle. This arm is here represented by dotted lines, since it is really in front of the other parts, and would cover up too much of them if shown in full. The knockoff cam C_1 and the safety cam C_2 are fastened to the cam ring E, which has a limited range of rotary movement, under the control of the governor.

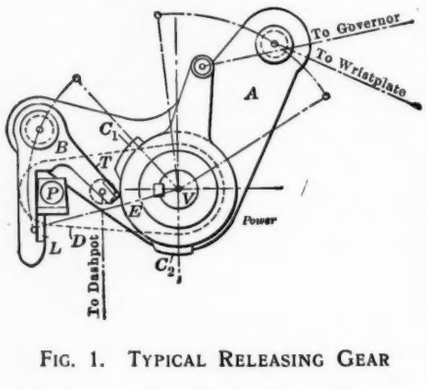


FIG. 1. TYPICAL RELEASING GEAR

In order that the valve shall be released, for quick closure by the dashpot, arm T must strike cam C_1 before lever A gets to its extreme of travel toward the right. In Fig. 2 is shown the condition in which, with pin B on crank A at its right-hand limit of swing, arm T just touches cam C_1; then the valve will not be released, but will come back slowly with the reverse movement of the wristplate, cutting off very late in the stroke. A single-eccentric gear is usually so proportioned that this condition will exist when the governor is down on the safety pin, so that when the speed is below normal due to a heavy overload or a bad drop in boiler pressure, the valve gear will work without release.

If the safety pin or stop is out, the governor can drop still lower, ring E turning still farther in a clockwise direction; then arm T will slide up on cam C_2 when arm B is in the position for engaging the valve (position marked B' in Fig. 2), and latch L will be held out so that it cannot hook up, and the engine will stop.

Now consider the eccentric setting and its effect on cutoff, as set forth in Figs. 3 and 4. The former shows the position of the crank and eccentric arms with the crank on dead center. The eccentric setting, defined by the angle COE or by the angle of advance GOE, is determined by the exhaust valve, so as to give proper release of the steam valve, and also compression. The particular angle on the diagram, $GOE = 15$ degrees, is a representative value for this type of gear, the advance being less than in a slide-valve engine.

Fig. 4 shows where the latest cutoff action by release of the valve can begin. The eccentric OE is still about 15 degrees above its dead center OB, but the wristplate is almost at the end of its travel. With the crank at 60 degrees as shown, the piston is at only one-quarter of the stroke; but the time it takes for the dashpot to pull the valve shut brings the actual valve closure to about 35 or 40 per cent. of the stroke. The range from this latest governor cutoff to the very late cutoff without release is entirely beyond the scope of utilization.

Suppose now that, as shown in Fig. 5, a separate eccentric OE_2 is used to operate the steam valves, leaving the exhaust valves to OE_1. This new eccentric can be set well back toward the crank; and when it gets near to dead-center position OE, Fig. 6 (the same as OE in Fig. 4), the crank will be at OC, corresponding to the position of the piston at perhaps five-eighths stroke, instead of at OC'. The 40 degrees, or so, of increased cutoff range is secured just where it will do the most good, for it is where the piston is covering the ground most rapidly. The angle $C'OC$, Fig. 6, gives the whole reason for adding a second eccentric to drive the steam valves, and shows this in about the clearest and most concise manner possible.

There is a limit to the amount by which the steam-valve eccentric may be set back toward the crank. If the negative angle of advance GOE, Fig. 7, is made too large, there will not be enough range of motion for the opening of the valve; that is, for bringing it from the rest position to the act of opening the port for admission of steam. Considering Fig. 7, we note first that the valve must be engaged or hooked up while the eccentric is near the dead-center line OA; and

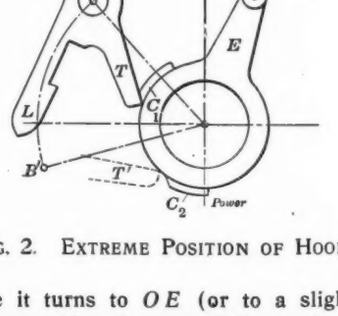


FIG. 2. EXTREME POSITION OF HOOK

while it turns to OE (or to a slightly earlier position, because of the lead) the valve is moved through its lap distance. The term lap is now used to denote the overlap of the closed valve. With a range AE or AF, as against AE' or AF' with the proper setting of a single eccentric, it is evident that the steam valve must have a small closure lap; but this may not be too small, lest leakage take place.

Giving to the word lap its usual meaning, or taking it to define the amount by which the edge of the valve overlaps the edge of the port when the eccentric is midway between its dead-center positions,

we see that a steam valve driven by an eccentric set as in Figs. 5, 6 and 7 must have negative lap. This means that if the valve were kept in continual connection with the wristplate it would close the port through much less than one-half of a revolution, and leave it open the rest of the time. Consequently, it is most essential that the condition of no release, represented in Fig. 2, shall be absolutely avoided with a two-eccentric gear.

In Fig. 8 it is shown that with admission taking place when the eccentric is at OE' (crank on or just below the dead-center line OC'), cutoff by a valve in connection with the wristplate will not occur until the eccentric reaches OE and the crank reaches OC. Since the exhaust valve is opened when the crank is above OB, it is evident that from this steam release down to OC there will be an open passage from the live-steam chamber to the exhaust pipe. With the proportions here used, live steam would pass through each end of the cylinder for about one-sixth of a revolution, or for about one-third of the total time, counting the two ends together. To make impossible this tremendously wasteful action, the requirement in terms of valve-gear design is that the cams C_1 and C_2 be set closer together on the ring, see Fig. 1, so that C_1 cannot go out of action before C_2 comes in.

In conclusion, it is appropriate to mention the scheme of increasing the cutoff range of a single-eccentric gear by giving the knockoff cam, corresponding to C_1 in Fig. 1, an oscillating movement so that even after arm A has begun to swing back from the position B, Fig. 2, the cam can follow up and overtake the trip arm T, releasing the valve. This idea has been applied on at least one design of Corliss engine for ordinary power service, and is in regular use on several makes of pumping engine.

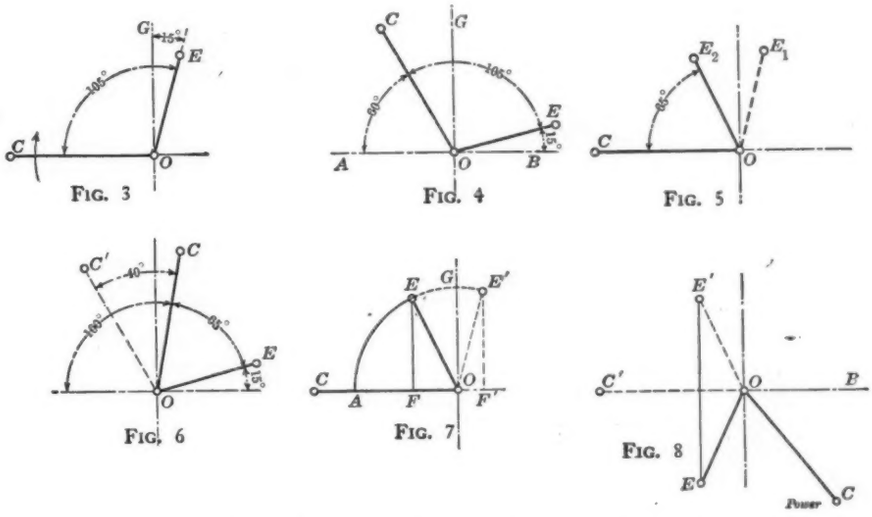


FIG. 3. FIG. 4. FIG. 5. FIG. 6. FIG. 7. FIG. 8.

EFFECTS ON CUTOFF OF VARIOUS ECCENTRIC SETTINGS

The Painting of Smokestacks

By J. C. Hawkins

Frequently the engineer is called upon to paint the smokestack or at least to oversee the job, so that the suggestions given by the author on how to do the work and the kind of paint to use will not be out of place.

Painting a smokestack is not usually considered a part of the engineer's duties, but he is frequently required to do so, or at least oversee the work, and a few hints along this line may not be out of place. The methods described in the following are used by several stack painters and "steeple jacks." Take, for an example, a steel stack 5 feet in diameter and 125 feet high, with two sets of guy ropes, one set 50 feet, and the other 90 feet, from the base, without a line through the pulley or hook at the top of the stack. First it is necessary to get together a suitable rigging consisting of one set of a single and double block with about 250 feet of ⅜-inch rope, and another set of blocks with about 500 feet of ⅝-inch rope, also two or three hooks made as shown in Fig. 1.

HOISTING THE FIRST ROPE

In order to get the first line to the top of the stack, a pole, long enough to reach from the roof to the first guy band, is procured and a hook similar to *B*, Fig. 1, which is made of ½-inch iron and has a spread of 4 inches. This hook is secured to the double block of the smaller set of falls at the ring. The hook is attached to the end of the tapered pole, as shown in Fig. 4, with cord or wire. The end of the pole is tapered to permit of removing it from the loop *C*. This hook is hoisted and hung over the first guy-wire eyebolt, close to the stack, and used to hoist the double block of the large falls into place. When the heavy blocks have been hoisted to the guy, the end of the small line is made fast at the base of the stack and the seat, Fig. 2, fastened to the single block. Then, with a short piece of ⅝-inch rope to tie up to the first guy and an extra hook, the painter is ready to hoist himself up.

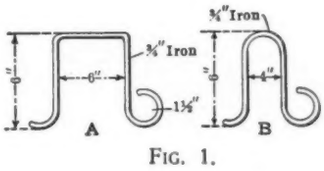


FIG. 1.

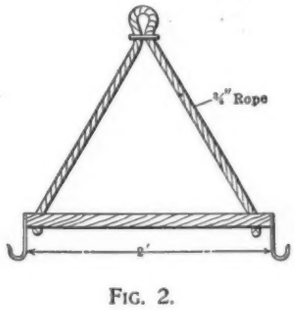


FIG. 2.

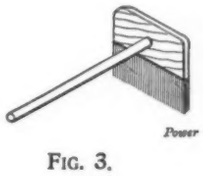


FIG. 3.

HOISTING THE FALLS

When the top block is reached the rope is made fast by bringing it under the seat sling and taking a hitch over the hook on the block, and then making a solid hitch with the short rope from the eyebolt to the seat sling, after which a helper can slack off on the small line. The pole is then prepared for reaching the second guy ring, as in the first instance. The painter steadies himself by pushing against the stack with his feet, and if, by any chance, he should get turned around, with his back to the stack, it is almost impossible to turn facing it again without a trip to the ground, or at best to the next lower guy.

After hooking over the first guy ropes, the painter reaches the top guy by pull-

ing the large blocks up and then pulling himself up, as before, from which position he can hook the pole over the top of the stack, using a hook like *A*, Fig. 1, and then proceed as before, to reach the top. Then by standing up on the seat and throwing one leg over the top of the stack, the double block of the large set of falls is hooked into the ring on the end of the hook as shown in Fig. 5, and the small falls are then lowered to the ground.

Painting the Stack

The painter is then ready for paint and brushes, which are pulled up with a small line. A 2½-gallon galvanized-iron or other light pail for paint will be required, also a short-handled brush, about 6 inches wide, and a long-handled whitewash brush such as Fig. 3 represents, with a 4-foot handle to which a wire loop is fastened on which to hang the brush when not in use. A scraper or large putty knife will also be needed. To do a good job it will be necessary to make three trips to the top, because, when swinging around, the painter can only get a stroke or two at a time. After painting as far as the reach permits, both ways, the painter lowers himself a few feet, and in this way gradually approaches the ground.

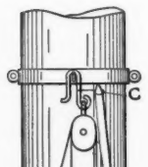


Fig. 4.

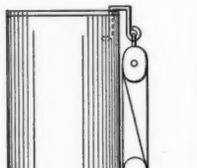


Fig. 5.

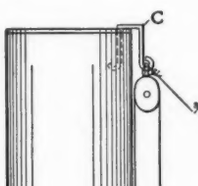


Fig. 6.

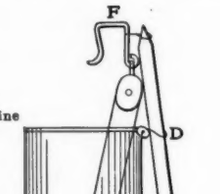


Fig. 7.

When passing the guys care should be taken to pass on the side which is to be painted next, and when the ground is reached the hook on the top of the stack can be worked around by giving the ropes a side swish.

Having painted the second strip from top to bottom the painter performs a trick which, if done rightly, will save an extra trip to the top; that is, he prepares the double block so that from the ground it can be unhooked from the hook over the top of the stack. To do this he passes the end of the ⅜-inch line through the eye of the hook and makes it fast to the block below the hook, as shown in Fig. 6, so that when pulling down on the ⅜-inch line the block is tripped out of the hook *C* and can then be lowered to the ground. This leaves the small line through the hook at the top of the stack. Fasten a ¼-inch wire rope, or large soft copper wire, to the end of the ⅜-inch line, by winding with twine, so that it will pass easily through the eye of the hook *C*, which is left on the stack for the next painting. It should be pulled tight and fastened to keep it from chafing against the guys and breaking.

A separate hook with a pulley fastened to it can be used to trip the large block, if desired, but this leaves two hooks on the top of the stack, and the one will work just about as well.

Another Method of Hoisting the Blocks

When there is a line through the hook at the top of the stack, the painter proceeds in a different manner to get the blocks to the top. A ¼-inch or ⅜-inch line is secured to the end of the wire rope and pulled through the hook, just the reverse of lowering the blocks. After getting the line through the hook, which may take some time and may break the wire rope or copper wire, as it will be worn thin in some places, a light pole, about 2 inches square and 12 feet long, is obtained and the ⅝-inch double block fastened to the tapered end of the pole, as shown in Fig. 7. Tie one end of the small line to the pole at *D*, about 3 or 4 feet from the top, and tie the other end to the bottom of the pole at *E*. The ⅝-inch blocks should be run out full length before hoisting. When the end of the pole, with the hook, reaches the top of the stack, a little twisting of the ropes will soon bring the hook *F* over the top of the stack, when the pole is lowered and pulled out of the loop and let down, leaving the block in place.

It would be easier to fasten the block to the end of the light line and pull it up, using the old hook and light line to work on, but it is not safe, as one cannot tell what condition the hook is in, and a good hook should always be used to work on.

Shifting from New to Old Hook

When the top is reached, if the old hook is found to be in good condition, the blocks may be shifted to it by using the short rope to hold the seat up to the first hook; then slack off on the blocks and transfer from one hook to the other. When leaving the top for the last time the small line is adjusted to trip the block out as in the case mentioned and the wire rope or copper wire is pulled up as before. Quite often the band around the top of the stack becomes loose, caused by the top of the stack eating away. This can be repaired by driving out the old rivets and replacing them with bolts with a large washer on each side, using clamps to hold the ring in place while putting in the bolts.

As this class of work is somewhat dangerous, the painter must personally see that all knots and ropes are right before trusting them, and he should have a good helper. A good ½-inch rope would, no doubt, be strong enough to hold the painter, but a ⅝-inch rope should be used as a measure of safety.

Paint to Use

Many use a paint made of coal tar, which is thinned with creosote or dead oil, but there are serious objections to this paint because the creosote and tar both have a tendency to pit or corrode the iron.

Before the stack is put up it should have two coats of red lead and linseed oil on the inside and a coat of graphite paint on the outside. The latter is a better preservative of the iron than tar paint but its disadvantage is its dull color; and if put on over tar paint it will peel off, as the oil works under the tar and loosens it. If put on often enough, however, until the tar is all off, it will give good service.

A good paint may be made by mixing flake graphite, linseed oil and lamp black together, the lamp black being used only as a coloring matter and the graphite as the body of the paint.

A good lubricant for hoisting ropes, according to *Mines and Minerals*, is made by mixing one bushel of freshly slaked lime to a barrel of coal tar, or a mixture of pure tar and tallow can be used. When pine tar which contains no acid is used as a base lime is unnecessary, as tar is solution proof to ordinary mine water. Another good mixture contains tar, summer oil, axle grease and a little pulverized mica, mixed to a consistency that will penetrate between the wires to the core and will not dry nor strip off. The lubricant should not be so thick as to prevent inspection of the rope, and after the first application should be used sparingly, so that the rope may be kept clean and free from grit. Graphite mixed with grease is another lubricating mixture that is used successfully, and sometimes pulverized asbestos is used instead of graphite.

The Design of Rope Drives

By Francis H. Davies

The influences of length and angle of drive, form of groove, ratio of pulley diameters, speed, size and character of rope, are outlined. Special types of drive, such as the cross, half-cross and angle, are considered.

Length and Angle of Drive

Of the many points to take into account in designing a rope drive the questions of length and angle are of high importance as both have a material influence upon the efficiency of the system. Upon the arc of contact which the ropes make with the pulleys depends the amount of power which it is possible to transmit. Failure to appreciate this fact is bound to lead to unsatisfactory results. Under any given set of conditions the longer the drive the greater will be the arc of contact, and in the case of horizontal working it is usually productive of the best results to have a distance between the shafts of at least three times the sum of the radii of the driving and driven pulleys. This will give an arc of contact of a reasonable number of degrees and will allow the ropes to be worked at their proper horsepower. A smaller arc will only deal effectively with a lower power. A drive which is too long is subject to serious jumping of the ropes under any sudden alteration of load, and to reduce this to a minimum it is necessary to increase the tension to a degree which, from the point of wear, both of the ropes and the bearings, is not desirable. Matters may be improved very considerably by the use of a grid or comb through which the ropes run, but, obviously, it is better practice to dispense with such adjuncts, if possible, and keep the length of the drive within reasonable limits.

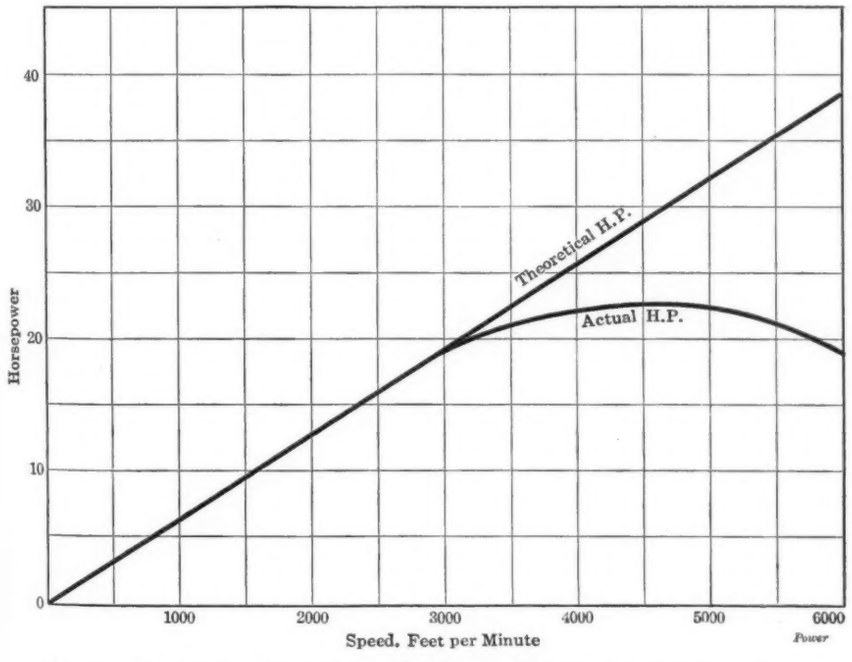


FIG. 1. EFFECT OF CENTRIFUGAL FORCE ON POWER TRANSMITTED BY 1¼-INCH ROPE

It is in horizontal running or at angles to the horizontal up to approximately 45 degrees, that rope driving gives the best results. In a vertical drive or one in which the angle is more obtuse than 45 degrees, the arc of contact becomes too low to give a good grip on the pulleys, and hence for any given power a greater number of ropes must be used. In a vertical arrangement, the power it is possible to transmit will be reduced about 25 per cent. from this cause. Where it is impossible to avoid the vertical or a very obtuse angle it is generally productive of better results if the bottom pulley acts as the driver, and under these conditions the single- or continuous-rope system is often preferred to the multiple drive as it is essential that the rope should be kept tight to prevent it hanging off the bottom pulley and the tension gear employed with a single rope is particularly useful in this respect.

Top or Bottom Drive

As a general rule it is preferable that the bottom should be the driving side, as under this arrangement the sag of the top side considerably amplifies the arc of contact. However, the reverse is sometimes purposely employed where, owing to a rapidly varying load, the ropes have a tendency to surge or jump. With the slack side at the bottom there is less likelihood of the ropes leaving the grooves, but they will not, of course, transmit the same power and a greater number must therefore be used.

Pulleys

Under ordinary conditions the pulleys should not be less in size than thirty times the diameter of the ropes. This is a general and safe rule, but circumstances occasionally necessitate some deviation from it. Where the pulley has to be of a smaller ratio the amount of power transmitted per rope must be reduced by the addition of other ropes to offset the excessive wear which takes place under such conditions. The more a rope is bent in passing over a pulley the greater is the internal friction set up between the strands and fibers, and the effect of this soon becomes apparent in the shape of wear. As the less bending a rope is subjected to the longer it will wear, it is always advisable to err on the side of large pulleys.

Grooves

The groove of which the sides form an angle with one another of 45 degrees has now become standard practice as it has been found to give the best results. It combines a maximum of frictional adhesion when the rope is on the pulley with a minimum of resistance when it leaves it, and does not tend unduly to convert the natural circular shape of the rope into that of a V. A more acute angle, while securing a better grip, wears the rope as it leaves the groove, and very shortly alters its shape in such a manner that its efficiency drops. An obtuse angle of 30 degrees or thereabout has little wearing effect, but it has been found by experience that it does not suffice to give the necessary grip. Some engineers prefer an angle of 40 degrees as there is then less tendency on the part of the rope to rotate around its center and so wear the surface; but the 45-degree groove with perfectly flat sides is by far the most usual and is, on the whole, considered the better. It is very difficult to assign a reason for the revolving of ropes as the phenomenon appears to follow no regular law. Generally speaking, it is deprecated, but there are not wanting engineers who prefer a certain amount of rotation. They contend that with rotation the wear is made more even.

It should be borne in mind that the rope should under no circumstances be

allowed to touch the bottom of the groove as this implies slip owing to the fact that under such conditions the rope cannot be wedged in the groove as it should be. Partly for this reason it is often advisable when ordering ropes to send a templet of the grooves to the rope maker, thus throwing upon him the responsibility for the success of that part of the drive. This is particularly the case where the grooves on the two pulleys differ in any respect as may at times be the case. So long as the difference is not enough to cause the rope to ride too high above the teeth in the one case and too near to the

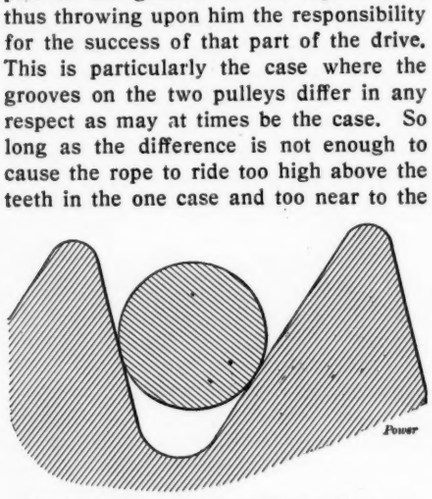


FIG. 2. GROOVE SUITABLE FOR OUT-OF-LINE DRIVE

bottom of the groove in the other there is no difficulty in working under such conditions.

NUMBER AND SIZE OF ROPES

Numerous tables have been published from time to time giving the horsepower that ropes will transmit at various speeds, and for ordinary conditions they are doubtless reliable. It must not be forgotten, however, that relative pulley diameter, length and angle of drive have an important bearing upon the power it is possible to transmit, and as before stated, it may be necessary under certain conditions to increase the number of ropes by 25 per cent. or more to insure freedom from trouble and undue wear.

In diameter ropes are seldom less than 1 inch, and owing to the resistance offered to bending and the consequent waste of power, a diameter of 2 inches may be taken as the other extreme. The most usual sizes are between 1¼ and 1¾ inches. These constitute a happy mean which gives a minimum of power loss by bending together with a maximum of weight for the production of the necessary amount of sag. Where the multiple-rope drive is in use, it is not correct practice to employ ropes of varying sizes on the same pulleys, as there is always a tendency on the part of the smaller ropes to run the pulley at a different speed, particularly where there is much difference between the diameters of the pulleys. This necessarily sets up friction and the smaller ropes will be found to have worn more quickly than the others.

There should always be plenty of clearance around a rope drive, not only to allow for a fair amount of sag but also for convenience in carrying out repairs. It is usual to assume that a horizontal drive under ordinary conditions will sag from 8 per cent. to 10 per cent. of the distance between the two shafts, and if the sag be greater than this, the ropes are running too slack.

SPEED AND HORSEPOWER OF ROPES

Most authorities agree that for main drives a speed ranging between 3600 and 5000 feet per minute is the best and most economical. It is quite possible to carry the speed much higher than this with safety, but, while some circumstances may justify it, as a general rule it is not good practice. Theoretically, the power it is possible to transmit with any given rope drive should increase in direct proportion to the speed at which it is run, but in practice this is by no means the case, as centrifugal force comes into play. This is demonstrated in Fig. 1. The straight line shows the power theoretically possible to transmit. The curve shows the manner in which the horsepower falls off above a certain critical speed between 4000 and 5000 feet per minute. In addition to this important consideration it is very necessary to remember that excessively high rope speed is inefficient in other directions. The life is materially shortened by the undue amount of bending which takes place, and air resistance to the movement of the ropes and pulleys becomes an appreciable factor. The horsepower which a rope will transmit with safety at ordinary speeds is very much in excess of that given in the average table, but it is not advisable greatly to exceed a load which gives a working strain of 200 pounds per square inch of sectional area as this means a correspondingly shortened life.

OUT-OF-LINE DRIVES

It happens occasionally that it is necessary to connect two shafts which are not strictly parallel. With ordinary pulleys there is an angle, depending to some extent upon speed, power, length of drive and relative diameters, at which the ropes will not keep in the grooves unless forcibly retained by means of jockey pulleys. This is usually put at anything over three degrees. It is unsafe, however, to place too great reliance upon a figure which may be a long way out under peculiar conditions. If such a drive cannot be avoided it is better practice to install pulleys having specially shaped grooves. A pulley groove which has been found very satisfactory is that shown in Fig. 2. While this groove retains the angular relation of 45 degrees between the two faces its center is several degrees out of the vertical, and the extent of this latter angle must depend upon the amount the two shafts are out of parallel. It is obvious that such an arrangement permits the rope to run on and off the pulleys without undue friction on one side and entirely obviates all risk of the rope mounting the groove.

CROSS AND HALF-CROSS DRIVES

The cross drive is best avoided where possible, but if it is installed in the right way it will, in most cases, be found satisfactory. The usual method is to employ a multiple-rope drive working in pairs, between each of which it is desirable to leave an empty groove. The ropes of each pair are crossed respectively right and left handedly so as to make the two center ones the driving side. The result is that as both are moving in the same direction there is no friction between them. The two ropes forming the outer or sagging side, having little strain upon them, are easily forced aside by the inner pair, and friction between the two although appreciable is not important. If it be desired to use one rope lapped twice around the pulleys instead of two separate ones, the tight side should be arranged to run on the outside, the sagging ropes crossing each other in the center.

In the half-crossed drive where one pulley is at right angles with the other it should be arranged so that the slack side runs on the straight and the tension side on the skew. The former will not enter the groove if the conditions are reversed, whereas the tight side does so readily.

RIGHT-ANGLE DRIVE

The right-angle drive is the least desirable of any and should be avoided if possible. The loss of power is neces-

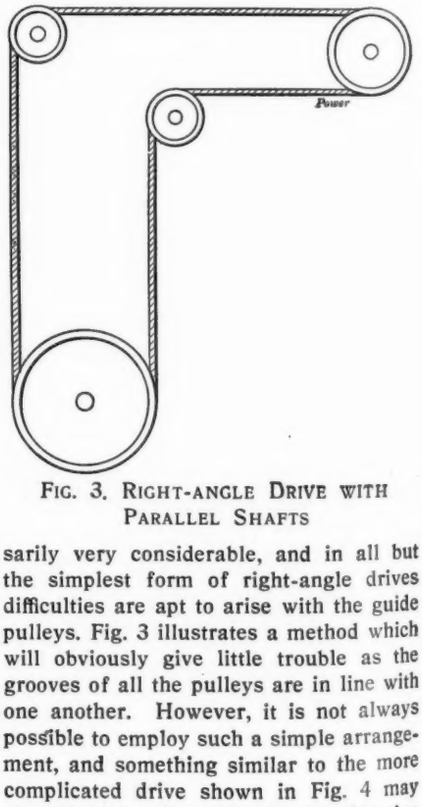


FIG. 3. RIGHT-ANGLE DRIVE WITH PARALLEL SHAFTS

sarily very considerable, and in all but the simplest form of right-angle drives difficulties are apt to arise with the guide pulleys. Fig. 3 illustrates a method which will obviously give little trouble as the grooves of all the pulleys are in line with one another. However, it is not always possible to employ such a simple arrangement, and something similar to the more complicated drive shown in Fig. 4 may be essential. A moment's consideration will show that under these conditions the grooves of the guide pulleys cannot all be in line with those of the driver and the driven; there is, therefore, a ten-

dency, on the part of the ropes to leave the grooves, and most careful design is necessary to secure satisfactory running. The design of the guide or jockey pulley will materially affect the operation of such a drive. Speaking generally, the grooves should be well rounded at the bottom and wide, but in complicated drives, particularly those of the right-angle class, it is often necessary to employ a special form which can only be arrived at after mature consideration of the particular problem. Grooveless guides, similar to a cotton reel in shape, have been tried and are as a matter of fact still frequently employed, but experience has shown that on the whole they are not desirable although there are certain simple cases which may be negotiated by such means.

Ropes

There is a certain amount of divergency of opinion as to whether the three- or four-strand rope is the better. It is claimed for the latter that it has four points of contact with the pulley as against three in the former, and for this reason it is supposed to give a better grip. Advocates of the three-strand rope advance the theory that the triangular form fits itself better to the groove, and

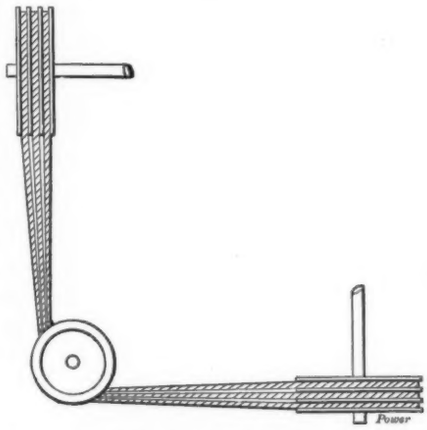


Fig. 4. Right-angle Drive with Shafts at Right Angles

a good deal of the preference shown for it is due to the fact that unlike the four-strand rope it possesses no core. Where first-class ropes are concerned it is possible to overestimate the importance of this feature. There is no denying the fact that in a badly made four-strand rope the core is very apt to take more than its fair share of the load and to fail prematurely; but it is certainly not fair to assume that all makes are alike in this respect, and looking at the matter from an impartial point of view there is probably little to choose between the two types of rope.

With regard to material, there can be no question that the cotton rope is better than the hempen. Owing partially to lower internal friction the life may be put at twice that of the latter, and there is less loss of power in bending, less stretch, and less noise. The cost is higher, but as a general rule it is worth incurring.

It is perhaps not generally known that in some branches of the rope trade it is customary to treat the rope with black lead or oil adulterants with a view of increasing the weight. This is sometimes carried to the extent of an addition in weight of 25 per cent. or even more, and since ropes are sold by weight, it is well to be alert to detect the practice.

A Convenient Form of Prony Brake

The prony brake shown in the accompanying photograph is used in the testing laboratories of Columbia University and has been found convenient for testing high-speed engines of relatively small powers; that is, up to forty or fifty horsepower.

The brake is very simple in construction, being made up of a coil of rope, a set of chainfalls, some lengths of pipe, a wooden beam and a platform scales. The pulley used in this case is carried in a ball-bearing hanger and is also double flanged, the outer flange keeping the rope from running off and the inner flange serving to hold the cooling water. It would not be practicable to use this style of brake with a plain pulley, as the rope would run off.

The chainfalls are mounted directly over the center of the pulley and the distance between the holes H, in the beam, through which the rope passes, is equal to the diameter of the pulley. The beam is suspended from the chainfalls by a knife-edge K, which allows free movement of the beam. The tension on the rear end of the rope balances the weight of the brake arm, thus making the reading on the scales direct, and avoiding the necessity of subtracting the weight of the brake arm.

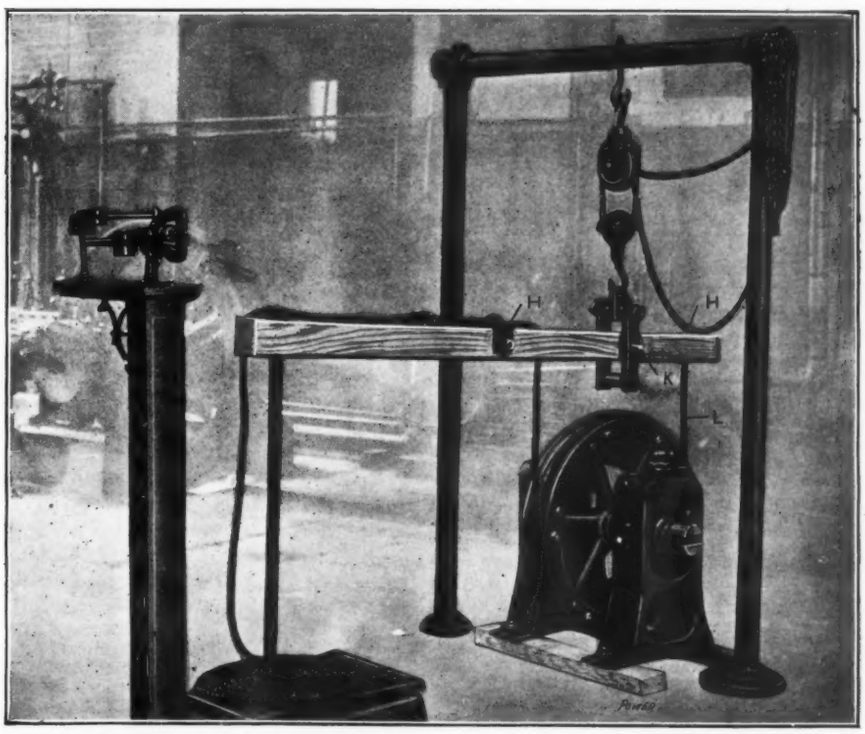
As the pulley is of the ball-bearing type, the pressure upon the bearings caused by taking up on the chain is negligible. After this tension has been adjusted approximately the scale beam may be kept floating by a slight tension applied to the rope through a piece of string, one end of which is tied to the rope at L and the other end of which is in the hands of the person making the test. The coupling at the end of the shaft serves to connect the apparatus with the engine to be tested.

The power is determined in the usual way, that is,

$$B.H.P. = \frac{2\pi \times W \times L \times N}{33{,}000}$$

where, $\pi = 3.1416$, W = scale reading, L = length of brake arm, in this case 4 feet, N = revolutions per minute. Hence, the formula may be expressed

$$B.H.P. = 0.0007616 \times W \times N$$



Prony Brake Rigged for Testing

Measuring the Flow of Large Streams

By Franklin Van Winkle

Of any stream the rate of flow is determined by multiplying the cross-sectional area by the average velocity of the water. It is not a difficult matter to compute the area, but the determination of the average velocity of current passing a given cross-section requires care and accuracy. The author illustrates plainly how to obtain the area and also the velocity of flow by means of surface, subsurface and rod floats.

In considering the value of a natural stream as a source of power or water supply, the volumes and variation in the flow available will be questioned with greater or less exactness. In estimating such a value it is unwise to form more than general opinion from average run-off conditions based upon drainage area tributary to the stream, or upon the record of other streams in the same locality, though a knowledge of those conditions may be very desirable. Direct observations should be made of the stream under consideration. The flow should be determined on the best information obtainable, and not only as to quantity of water available under maximum and minimum conditions, but also as to duration and extent of fluctuations between these limits.

Although average and extreme low-water conditions generally control or limit the extent and usefulness of a plant, a knowledge of the volume of flood flow is of importance in arrangement and construction of dams, designing capacities of spillways and in making provision against floor damage to buildings and other works. These considerations of flow of a given stream are only determinable from actual tests.

It is not expected, however, that one should delay his decision as to character and extent of a development until a series of tests have actually been made covering every stage of flow worthy of his attention. Inspection of the banks, high- and low-water marks, investigation of the uses made of waters of the same stream above and below the site in question, and testimony of the younger as well as that of "the oldest inhabitants" of the locality, will serve as valuable checks on opinions formed from actual tests and estimates of flow made at different stages of stream flow.

In a previous article* it was stated that under most circumstances, the measurement of a quantity of water could be determined by ascertaining the velocity of its discharge through some kind of passageway or orifice whose dimensions were ascertainable. For most practical purposes, measurement of the flow of streams can be satisfactorily performed by direct determination of velocity and cross-sectional area of the natural channel, and for measurement of flow of large streams it is usually the case that no other methods are practicable.

As rate of flow consists of the product of a given cross-sectional area and the average of velocities of all different portions of the stream as it passes the given section, the value of a computation of flow depends directly upon the care and accuracy with which both of these factors have been obtained. Measurement of cross-sectional area may present no unusual difficulties, but determination of average velocity of current passing a given cross-section is difficult from the fact that the velocity of the stream at every point of the cross-section is continually changing.

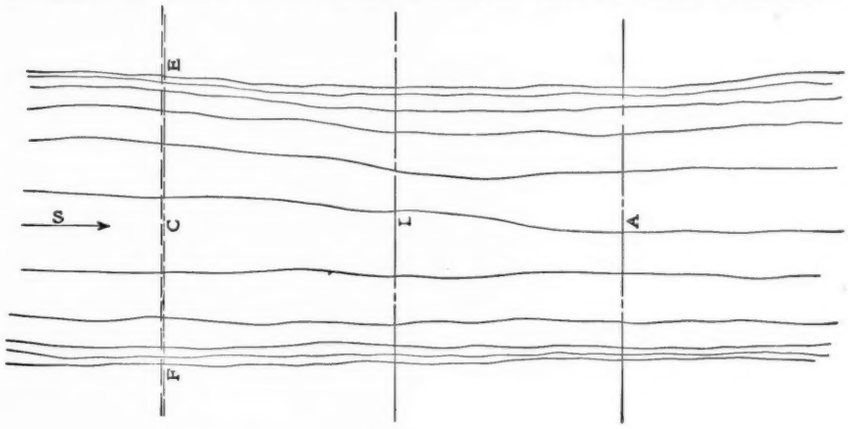
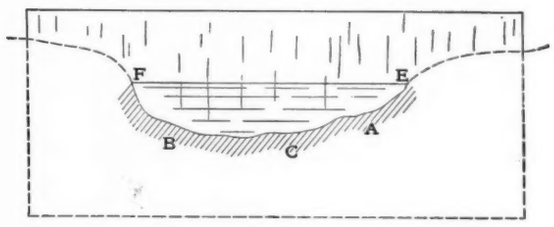


FIG. 1



Power, N.Y.

FIG. 2

Erratic Motions of Flowing Water

The irregular rolling, surging and boiling motions of mountain streams, so clearly indicative of continuous changes in direction and velocity at every point of a cross-section, continue, in some degree, even after the waters reach a point down stream where the flow is smooth, noiseless and with velocity that is scarcely perceptible. The motions of flowing water are little less erratic than those of the wind. By releasing a small quantity of heavy oil, liquid dye or a handful of bran some distance below the surface of a quietly running stream of clear water, these irregularities of currents and velocities are made apparent by their drawing out the discolorations in a confusion of intertwining figures, and great irregularities of flow exist even when the rate or discharge of the stream may be practically constant. They seem to be largely dependent upon the alinement, length, shape and roughness of the channel, but are present to greater or less degree in streams flowing in all kinds of channels. Open-stream measurements should, therefore, be made with due regard to the existence of continuous fluctuations in direction and of velocity of currents, and methods should be employed which will result in elimination of errors compatible with the purposes of the gagings.

Methods of Measuring Stream Flow

When a stream is large, or for other

*Power, August 25, 1908, page 313.

reasons employment of weir measurements would be impracticable, recourse may be had to other direct measurements of flow which, in many instances, are capable of being employed with as much exactness as the occasion will demand, and without being attended by the loss of time, trouble and expense incident to installation of a weir.

In considering the merits and adaptability of any method the fact should not be lost sight of that employment of natural cross-section of the stream is common to all; that the measurement of cross-section and of velocities constitutes the main factors in computation of stream flow, and of these two factors, accurate measurement of velocity is the more difficult to obtain. Any of the methods which may be employed require accurate determination of cross-sectional area of the stream at one or more places, and any measurements made for that purpose should be taken with great care. The means employed must be adapted to the size of stream and the facilities afforded for the work. The detail of processes employed by the engineer or surveyor may differ from those used by the mechanic, but each having the same object in view should arrive at the same result.

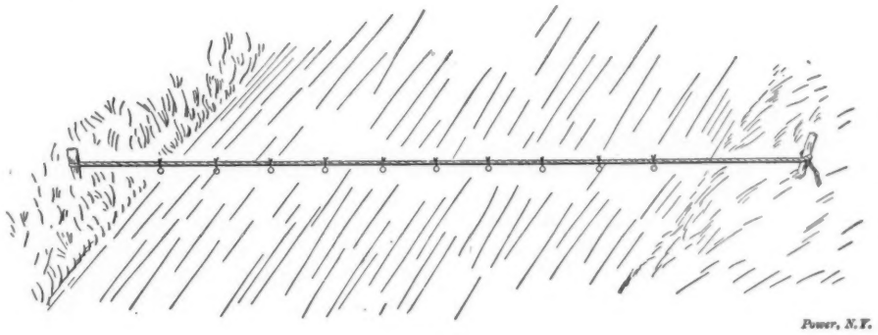


FIG. 3

CROSS-SECTION OF THE STREAM

Ordinarily the cross-section is understood to be in a vertical plane at right angles with the thread or main direction of the stream as it passes the place where the section is taken. Supposing that cross-sections are to be taken of the stream at places *A*, *B* or *C*, Fig. 1, a representation of the cross-section taken at any point, as *C*, would present the same appearance as though the waters of the stream, without changing their hight, had suddenly come to rest, and then cutting the stream crosswise by insertion of a vertical plate of glass like a dam. The outline of water would present an appearance similar to Fig. 2 in which the section of the surface of the stream from *F* across to *E* in Fig. 1 would be represented by a horizontal straight line, like *FE* in Fig. 2, and the irregular shaded line *FBCAE*, Fig. 2, would represent the cross-section of the channel; and the whole figure which is contained between those lines, would be the cross-sectional area of the stream taken at the point *C*, Fig. 1.

The purpose of taking cross-sectional measurements is to determine the form and area of cross-section of the stream, for if this area, found in square feet, is multiplied by the average velocity in feet per second at which all parts of the stream flow through such a section, the product will be the whole volume of stream flow in cubic feet per second. Determination of the form and magnitude of a section will require measurement of breadth of the stream, and a number of measurements of its depth, taken at stated intervals, across the whole width at the place where the section is taken. The closer the intervals of depth measurements, the more accurately will the section be obtained, but for convenience in computation of the sectional area it will be best to take depth measurements at equal intervals.

OBTAINING THE CROSS-SECTIONAL AREA

For obtaining sectional areas of ordinary mill streams, good results are to be obtained by employment of very simple instruments. Depth measurements can be taken with any convenient form of measuring rod, taking care, however, that it is not pushed down into the bottom when taking the readings. To prevent the rod from penetrating the bottom, its lower end should be provided with a plate or disk several inches in diameter. A length of ¾-inch iron pipe makes an excellent measuring rod if provided with a half-flange fitting screwed on the lower end, and for extensive measurements it will pay to lay off the rod in feet, with each alternate space, painted red and white like a surveyor's range pole. Intermediate divisions of inches or tenths of feet can also be laid off, or measurements of the fractional parts of feet can be measured as taken with a 2-foot rule. For measuring depths up to 3 or 4 feet, the writer prefers a newly planed rod of white pine, about ⅞x3 inches, on which to mark each depth and its number with a lead pencil on the side of the rod, subsequently measuring and recording results. By this method measurements are more quickly made, and the same person making the measurements may record them with much greater convenience than when continuously handling a rod in and out of water. For obtaining proper spacings of depth measurements across a stream, one end of a tape line can be secured to one bank while an assistant draws the tape to the other bank, and then by wading along the down-stream side of the tape line, or rowed in a boat by another assistant, measurements of depth can be taken at such intervals as may be desired. In the absence of means for holding a tape line in place, the intervals can be laid off with small tags spaced at equal distances along a stout cord or small rope when drawn taut, and this can be stretched across the stream suspended from trees or stakes, as shown in Fig. 3; but in any event exact measurement should be made of the whole width of the stream.

No matter what system of recording dimensions may be adopted, before leaving the site a good free-hand sketch should be made of the section with all dimensions inserted, as shown by Fig. 4. Paper with square rulings or ordinary section paper is best for this purpose, as by their aid free-hand sketches may be drawn very accurately to scale and checked before leaving the location. For most practical purposes, depth measurements taken at intervals of about 2 feet apart will be satisfactory, but in any case, to avoid confusion, the intervals should be equal. When any number of equally spaced depths have been taken, the area of the section is readily found by mutiplying the breadth by the average depth, but when the section of the bottom is very irregular, as shown in Fig. 5, it will be more accurate to make computation of the whole area from a careful platting of the section.

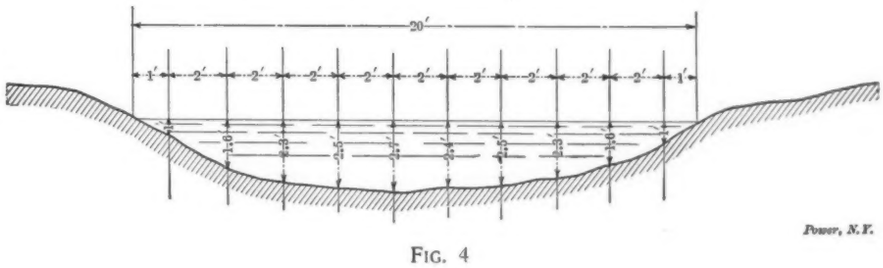


FIG. 4

The cross-sections of large and rapid streams are obtained by soundings taken in the manner illustrated in Fig. 6. Sup-

posing *A B* to be the line on which soundings are required, a base line *C D* is measured off along shore, preferably, though not necessarily, at right angles to *C D* and range poles are set at *C* and *B*. A boat is allowed to drop down

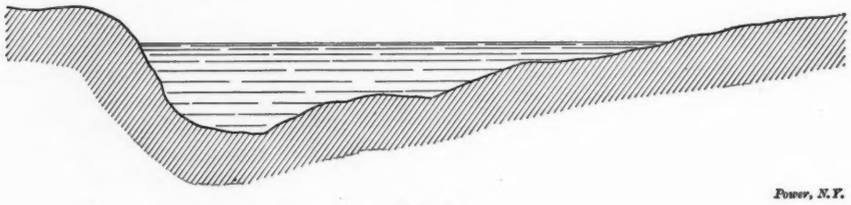


FIG. 5

stream from a point above *E* and when it comes in line with the range poles *C* and *B*, at a signal from a person stationed at *A* the lead is dropped and a person stationed at *D* observes the angle *C D E* with a compass, transit or theodolite. The sounding line may have a weight of about 14 pounds of lead, and if the boat drops down stream slowly, the weight may be allowed to hang near the bottom,

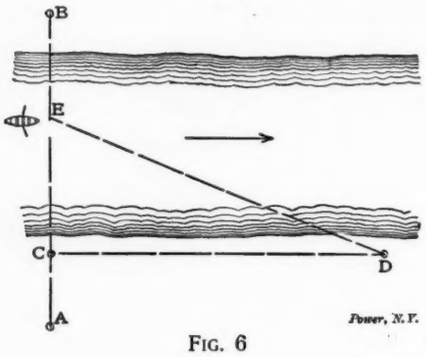


FIG. 6

so that there is no loss of time in taking the sounding when it arrives over the line *C B*. In place of using a theodolite or other angle-measuring instrument, the observer at *D* may be provided with a "plane table" and simple form of alidade, as shown in Fig. 7, with which to observe and lay off different angles. In extensive surveys of large rivers, observers with theodolites are stationed at both points *C* and *D*, the instrument at *C* being directed across the stream on the line *A B*, while that at *D* is kept trained on the boat. When the boat arrives on the line *A B*, the observer at *C* signals, the sounding line is dropped and the observer at *D* reads off the angle *C D E*. By setting up range poles or other conspicuous markers at points *A*, *C* and *D*, the use of an angle measurement made by a person stationed at *D* and another stationed on the line at *C* can be dispensed with by taking readings of angles subtended at *C E D* with a box sextant carried by a single observer in the boat.

By repeating observations taken at different distances from shore, the location of different soundings can be platted in their proper position, and the form of the river bed drawn by connecting the extremes of the lines from which the average section is readily determined.

MEASURING VELOCITY

As velocities at different points of a cross-section are variable, average velocity is only an imaginary one which if it existed in every point of the cross-section, the discharge would be the same as that resulting from the aggregation of filaments with uniform velocity. The least error in obtaining average velocity by any method may therefore be expected when gagings are made in channels which are

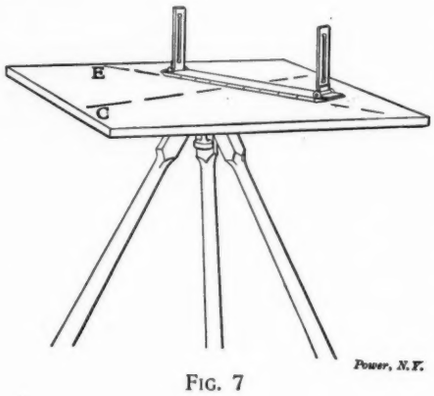


FIG. 7

straight, smooth and uniform in cross-section and the flow is free from swirls and eddies.

Methods of measuring velocities may be divided into three general classes:

(1) By speed of floats passing over a measured length of channel.

(2) By instruments for measuring velocities of different points of a given section.

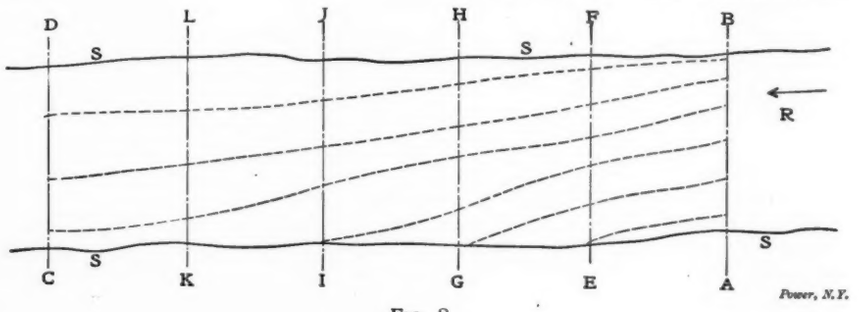


FIG. 8

(3) By inference of velocity from temperature due motion; or measurement of inclination of the surface of a stream, combined with dimensions of its cross-section and assumed roughness of the channel.

Measurements made by floats are the simplest, requiring only the observation of the time it takes for a floating body to move over a measured course; while measurement of velocity by instruments requires that a constant for the instrument must first be found by experiment. Methods coming under the third head are almost wholly theoretical and will hereafter be referred to only for the sake of greater completeness in treating this subject. Under favorable circumstances, measurements by floats and by instruments can be made to yield results which for accuracy closely rival those obtained by weir measurements. Each has its advantages and disadvantages.

When measurement of flow is based on the velocity of floats passing over a measured distance, then it is necessary to arrive at an average value of cross-sectional area of the stream for the spaces passed over by the floats. For this purpose three or more equally spaced cross-sections must be obtained of the measured course, as at *E F*, *G H*, etc., Fig. 8; and for obtaining the flow in cubic feet per second, the average area of cross-section in square feet is to be multiplied by the average velocity of the stream in feet per second over the measured distance.

By the float method the mean velocity of only comparatively few particles of water is obtained for the time required for the float to move from one point of observation to the other, i.e., a float is acted upon only by the particles near it during the time of its passage over the measured distance, for it is carried along by a single impulse of the water and is of no service in indicating velocity due to succeeding impulses, which may be greater or less than the impulse which moves the float. On the other hand, instruments stationed in a given cross-section receive and measure the velocity of succeeding particles that strike against them during the observation. If there were no irregularities in velocities and particles followed one another past a given point with uniform velocity, then, in absence of wind or other disturbing elements tending to divert a float from its proper course, the float method of measurement would be all that could be desired. But since these irregularities of velocity exist in all moving waters it

is believed that measurement of velocities of streams, wherever practicable, by employment of velocimeters and measurement of a single cross-section is preferable to float measurements of velocities combined with the average of several sections.

Surface Floats

Any small light bodies, such as small pieces of wood, wax or other material which will float and can be readily observed from the banks can be used as surface floats for measuring the surface velocity of a stream. Small dependence can be placed on floats, however, unless the course selected is over a fairly straight portion of channel, uniform in breadth and depth, and there is no disturbance from wind, eddies or other causes tending to separate the float from the same surounding particles of water in its passage from one end of the measured course to the other. The measured course should be 50 to 200 feet long. The floats should be dropped in the stream some considerable distance above the beginning of the measured distance so as to take up the velocity of the stream before traversing any portion of the measured distance.

In Fig. 8, *SSSS* representing the banks of a stream flowing in the direction indicated by the arrow *R*, suppose the distance *A C* to equal 200 feet, to be laid off on the left bank of the stream. If the stream is narrow enough some kind of lines bearing equal-spaced tags, Fig. 3, or equally divided rods, should be stretched from bank to bank at *A B* and *C D*, dividing the breadth at the beginning and end of the course into five or more equal spaces. The floats released at *R* should be paced at such distance from the banks as to cause the passage of one after another under different divisions of the line *A B*. An observer stationed at *A* should note the space under which each float crosses under the line *A B*, noting the time. To avoid confusion, a second float should not be released until the preceding one has passed under the line *C D*. The time required for passage and the space of *C D* passed under should both be noted. Intermediate lines *E F*, *G H*, *I J* and *K L*, equally spaced apart and corresponding with *A B* and *C D* in divisions of tags, will serve in obtaining a record of intermediate velocities and of the courses taken by the floats. It very commonly happens that many of the floats started under the beginning line *A B* are stranded at one side or the other of the channel before reaching the end of the measured course, and if their paths were to be platted the courses of the floats would be represented by the dotted lines shown in Figs. 8 and 9. In such cases, little more can be done than to take into consideration a shorter length of measured course, first taking into account only the time of passage of equally spaced paths of floats which have traversed the measured distance which is taken, and then, assuming that the average distance traveled by the others is only one-half as far in the same time, the time required for their passage over the whole measured distance may be regarded as twice the average time required for passage of those which do pass over the whole measured distance. Suppose, for instance, that the measured distance taken is *A C*, Fig. 9. The paths of floats 1*p*, 2*p* and 3*p* are terminated before reaching the section of the stream *C D*. If the average time required for passage of floats by the remaining paths 4*p*, 5*p*, etc., over the whole measured distance *A C* was 40 seconds, then in making up the whole average time, the time attributable to floats describing paths 1*p*, 2*p* and 3*p* should be assumed as 80 seconds.

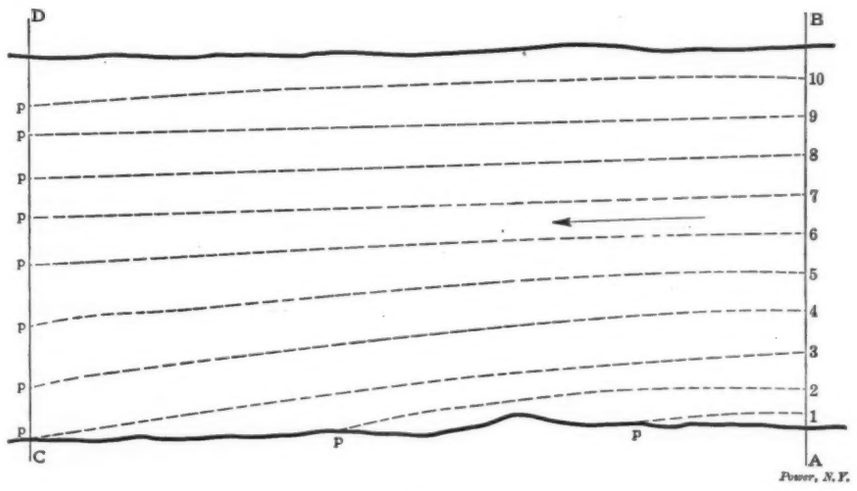


Fig. 9

Dividing the measured distance *A C* in feet by the average time in seconds thus found, gives the average surface velocity in feet per second. Having obtained the average surface velocity, the mean velocity of the stream must be determined from its relation to surface velocity. This is by no means uniform, nor can it be said that it is well established. Mean velocities are frequenty assumed to be about 80 per cent. of surface velocities, but careful experiments have shown that the relation between surface and mean velocity is only a chance one, depending on proportions of length, breadth, depth and shape of channel and is constantly varying. Inspection of Fig. 9 with respect to such paths as 1*p*, 2*p* and 3*p* suggests that the surface velocities along those paths have been arrested but must exert reactions on the mean velocity which are impossible of analysis.

In his "Lowell Hydraulic Experiments," Mr. Francis reports that surface floats of wax, about 2 inches in diameter, passing down the center of a long rectangular flume 10 feet wide by 8 feet deep moved about 6 per cent. slower than a tin tube 2 inches in diameter which extended a few inches above the surface and was loaded so as to stand nearly vertical, reaching within a few inches of the bottom of the flume. Also that the velocity of the tube was less than the mean velocity of the water in the flume. In another rectangular flume, which was 20 feet wide by 8 feet deep, he reports that the velocities of the tubes were about ½ per cent. greater than the average velocity of the entire body of water; and in a flume 29 feet wide by 8 feet deep the velocities of the float proved to be about 10 per cent. greater than that of the body of water. Charles Ellet, Jr., in tests of velocity of the Mississippi, concluded that the mean velocity was greater than the surface velocity by about 2 per cent., but in shallower streams he always found surface floats to travel faster than subsurface floats.

These examples are quoted to show the difficulty in assigning any particular relation between surface velocity and mean velocity, and although the surface float has frequently been relied upon in important gagings, its employment would appear to be misleading rather than useful.

Subsurface Floats

The defects of surface floats in indicating only surface velocity are largely overcome by employment of submerged floats, though the latter cannot be employed in places where grass, weeds or other obstructions occupy portions of a channel where it is desired to obtain velocities. For measurement of flow in large open channels, subsurface floats have been used with excellent results. The same general methods are employed for obtaining the mean velocity as with surface floats, excepting that the indicated average velocity is assumed to be the mean velocity from the fact that the largest portion of the float is well below the surface, and the float as a whole is assumed to partake of the mean velocity of adjacent vertical films of water by which it is carried down stream.

The double float consists of a light surface float and a submerged float, the latter somewhat heavier than water and usually connected to the surface float by a cord or wire, the purpose of the upper float being to support enough weight of the lower float to keep the latter at a constant depth and indicate its position. The connecting cord can be lengthened at will so as to permit the lower float to be carried along by the velocity of the water at different depths.

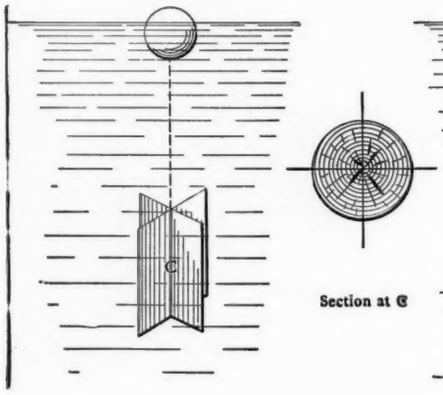


FIG. 10

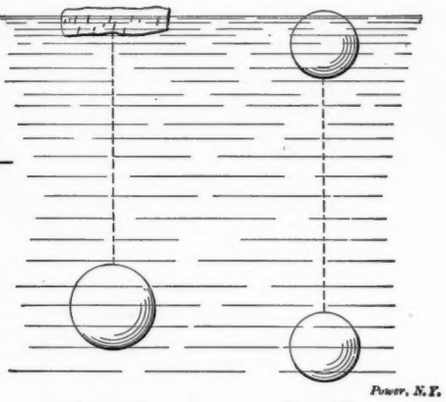


FIG. 11

FIG. 12

Fig. 10 shows a form of subsurface float which has frequently been employed and recommends itself for economy of construction. It consists of two tin plates bent at right angles and joined together at the angle by soldering, and connected by a cord to a small wooden ball for a surface float. In place of soldering the parts of the lower float, they may, to good advantage, be secured to a piece of wood set in the angle as shown in the sectional view. Fig. 11 shows a form of submerged float consisting of a hollow metal ball connected to a piece of cork which acts as a surface float. Fig. 12 illustrates twin floats consisting of equal-sized hollow spheres, the upper one being a little lighter and the lower one a little heavier than water. According to Captain Cunningham, who made extensive gagings of natural streams for the British Government, the twin float gave better results than ordinary forms of subsurface floats, being less disturbed by wind and waves; but being almost completely submerged they are more difficult to locate.

Fig. 13 is an illustration of a subsurface float such as used by T. E. Ellis in the Connecticut river survey (1874). The submerged portion *A* consisted of a hollow annulus of tin plate 8½ inches in diameter by 8½ inches high and 7½ inches inside diameter, which, sealed at both ends, provided a ½-inch air space all around the cylindrical portion. Two brass wires were soldered across the lower end at right angles to each other, and to these a 28-ounce lead sinker was attached; two other wires were soldered at right angles to each other at the center, coming together near the upper end of the annulus, and to these the connecting cord was attached. The surface float *B* was a hollow ellipsoid made of tin plate, 6 inches in diameter and 1½ inches deep, with a cork in the top to hold a small flag and a small eye in the bottom for attaching the connecting cord.

Double floats of the same general description were used by Humphreys and Abbott in gaging the Mississippi river (1851), and the same method of gaging that river and its tributaries was used almost exclusively by engineers of the Federal Government from 1851 to 1881, but since the latter time the use of floats has been largely superseded by employment of improved forms of current meters.

Employment of double floats was found by Messrs. Humphreys and Abbott to be the best method available to them for obtaining reliable results and as their gagings were conducted with great care,

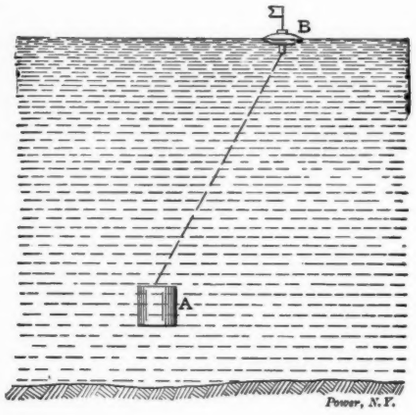


FIG. 13

brief description of their manner of using double floats, illustrated by Fig. 14, should be of interest. A suitable place was selected on the river, a base line *A B* 200 feet long was measured off on one bank parallel to the axis of the current, and sections *A A'* and *B B'* at each end of this base were marked out at right angles to it. The time consumed by each float in passing between these sections and the position of each float when it passed them were noted by four men, two at each end of the base, each party having a stop watch and a theodolite. At a signal of the engineer in charge, the floats were placed in the river from a boat about 100 feet above the upper section *A A'* and after passage over the measured course were picked up by a man in a boat some distance below the lower section.

At the instant a float passed the upper section a signal was given, the watches were started, and the angular position of the float was read with both theodolites. The instant the float passed the lower section a signal was again given, the watches were stopped, and the angular position was again read with both theodolites. These readings gave the distance of the float from the base line when it passed the sections and two independent measurements of the time consumed by it in passing between the sections. Soundings were taken at the end sections and at one or more intermediate sections, from which the area of the mean section was computed.

With this method base lines over 300 feet in length have been employed. In deep rivers, or in streams carrying weeds and grass, measurement of velocity by means of floats is about the only one which is practicable. It is impossible, however, to determine the position or depth of the lower float. Its position is approximately indicated by that of the upper float, but its relative position varies with the direction and velocity of the wind and the length of cord connecting the floats, and on account of side pressure on a long connecting cord combined with the vertical upward or downward action of currents on the lower float, it is difficult to determine the depth of the lower float with precision. The upper float may drag the lower one or *vice versa*. Employment of the double float is, however, very superior to that of the single surface float in obtaining mean velocity.

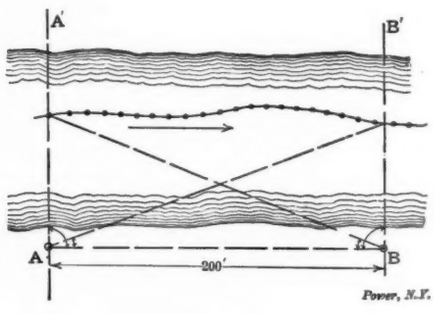


FIG. 14

ROD FLOATS

Rod floats have been employed for measuring mean velocity of flowing streams for many centuries, the earliest experiments recorded of their use for

this purpose being those of Professor Cabeo, in 1646. They may be made of wood or thin metallic tubing, from 1 to 2 inches in diameter, weighted at their lower end so as to float nearly vertical. The lower end should nearly touch the bottom and the upper end should rise a few inches above the surface and may be provided with a small flag or tuft of cotton at the upper end so as to be readily located. One of the best forms of float rod consists, as shown in Fig. 14, of a wooden rod with a metal cap at the lower end in which to place different amounts of shot for sinking the rod to proper depth. The rod can be made in several sections of length to be screwed together according to depth of stream where it is employed. For measuring velocities of raceways 2 to 3 feet deep, an ordinary broom handle, loaded at one end, makes a very good rod float.

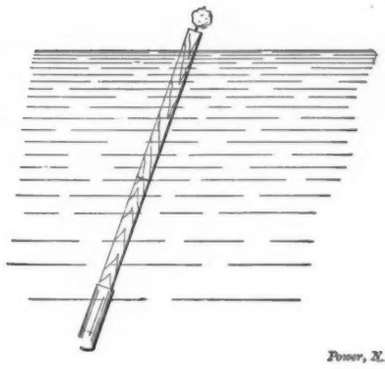


FIG. 15

Rod floats are to be employed for obtaining average velocity in the same general manner as that for using double floats. In obtaining the flow, a measured distance should be carefully measured off, the average cross-sectional area of the stream for the measured distance should be obtained with accuracy, and this combined with the average velocity of a number of floats released at equal distances across the breadth of the stream from bank to bank. For canals and raceways of moderate and uniform depth and for gaging streams containing floating grass and weeds, this is undoubtedly the best method that can be used. While float rods cannot be used for obtaining average velocity in very deep streams or in streams of rough or irregular bed, and may be slightly affected by wind, they interfere but very little with the natural motion of the water, they measure velocity direct, are not affected by silt and weeds, they measure forward velocity, are readily obtained and are inexpensive.

Quick Repair of Compressor Cylinder

By A. J. Dixon

A refrigerating apparatus of the absorption type, used to cool drinking water for a public building, was put out of service in the midst of a spell of torrid weather by an unusually severe leak in the piston-rod stuffing box on the ammonia cylinder of the pump which returned the liquid from the absorber to the generator. Inspection showed that the leak was caused by a crack extending longitudinally through the wall of the stuffing box on its under side, for about 8 inches from the end, as shown in Fig. 1.

Midway of the length of the stuffing box, which was about 14 inches overall, a cast-iron gland was placed between the rings of fibrous packing to entrap any leakage from the cylinder past the inner section of the packing and to return it

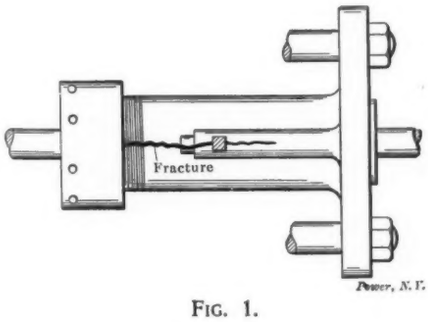


FIG. 1.

through a duct on the under side of the stuffing box to the suction chamber. This suction duct was formed by drilling two holes at right angles through a lug cast on the barrel of the stuffing box, the holes being closed on the outside with ⅛-inch pipe plugs. These details are all shown in Fig. 2.

Regarding the origin of the fracture, the most plausible explanation seemed to be that in operating the pump the stuffing-box nut and gland had been drawn up so tightly as to compress the packing longitudinally and expand it radially, creating an excessive tensile stress in the wall of

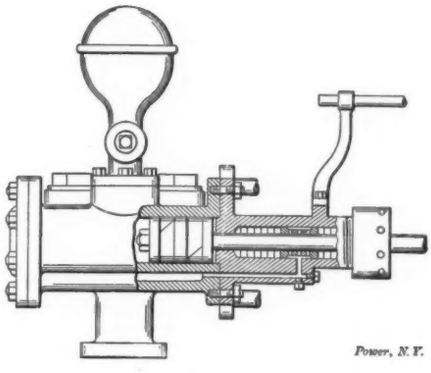


FIG. 2.

the stuffing box, thus starting a small crack which ended in complete rupture through the line of drilled holes, this being the weakest longitudinal section of the stuffing-box wall.

The substitution of a new cylinder head and stuffing box would have necessitated suspending cooling operations for several days, but this being intolerable, considering the weather, the engineer was constrained to devise an immediate repair. Several schemes were considered, but the plan finally adopted is shown in Fig. 3.

The cylinder head, with which the stuffing box formed a single casting, was first clamped to the faceplate of a lathe and the thread for receiving the packing nut was chased back to the boss which formed the support for the valve-stem bracket. An extra-heavy blind pipe flange, 10 inches in diameter, was then bored and threaded to a force fit upon the stuffing-box thread, and with a 12-foot lever and three men, whose weight aggregated 580 pounds, the flange was run onto the position indicated in the sketch. This accomplished, the pump was reassembled and again put to work under the usual discharge pressure of 150 pounds per square inch, without showing the semblance of a leak.

The condition of the fracture indicated that during its development, particles of

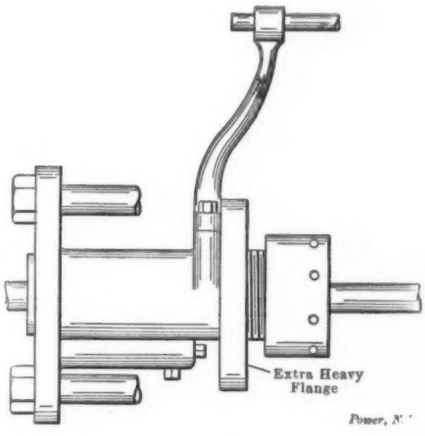


FIG. 3.

the rod packing had been gradually squeezed into it, and these fibers serving as packing for the joint, assured an absolutely tight job when the edges of the fissure were drawn together under the pressure of the band formed by the pipe flange.

The Saving of Heat Units*

By H. L. Peiler

Small leaks, which individually appear insignificant, in the aggregate make a sum total at the end of the year that is surprising. Discharging condensation to the sewer is a leak of no small importance. All steam-using machinery should be amply and quickly drained, and the water of condensation kept under pressure and returned direct to the boilers without loss of its heat units.

On going over a certain plant with the owner, I asked him where a particular return line led that originated in a dry room. He replied that it emptied into the drain. I asked how much water he was losing, and he stated that it could not be much and that the water was cheap. However, I managed to arouse his curiosity sufficiently to have the pipe uncovered and the water measured. To his surprise he found that a 50-gallon barrel was filled every 40 minutes and that the temperature of the water near the outlet of this large drier and while still under pressure was 306 degrees Fahrenheit. His engineer calculated for him the approximate cost, in tons of coal, of heating nearly 250,000 gallons of water from 50 to 300 degrees Fahrenheit, and this manufacturer came to the conclusion that he was not sufficiently wealthy to continue that system, when at relatively low cost the same water could be returned to the boilers direct and without loss of heat units. This plant had almost every practical device for making steam cheaply. High-class boilers, modern engines, approved equipment everywhere, including ball-bearing shafting and everything to reduce friction loss to a minimum, and yet this was only one of a series of "leaks" that were discovered during a careful investigation, and eventually he found that leaks, which cost him $1800 to stop, had for years previously been trickling $2600 annually into the drains. Everything possible had apparently been done to make the whole plant modern and uptodate, which is equivalent to saying "economical to operate," but the men in charge had neglected to "look into the furrows." This is only one of a series of similar incidents which I can recall in my own experience.

For purpose of comparison, it is necessary for me to state briefly the conditions that are most frequently met with in steam-using plants in this country. They are, of course, quite familiar to all engineers. The steam is trapped at the point of discharge at the apparatus in in which it is used. The water is then usually run by gravity to some low point, gathered in an open heater, vented receiver or hotwell, and after the make-up water has been added, the boiler is fed from this central reservoir. Where the steam is used under very low pressure (zero to five pounds) and where exhaust steam from the engines is mixed with this low-pressure live steam, the temperature of this water of condensation upon reaching the receiver is frequently under the boiling point and the addition of the cold make-up water further reduces it to a point where it can readily be handled by a good hot-water pump. Under these conditions, it is not uneconomical to handle the returns in this manner. But here I would like to point out that in drawing hot water from an open heater or receiver, and passing it through a pump, the temperature is still further reduced, and many engineers who have thermometers on the heater or receiver overlook the important fact that in most cases they lose from 8 to 12 degrees between the heater and the point where the feed water enters the boilers, and that this means a direct loss of from ⅔ to over 1 per cent. of the total fuel consumption.

However, there are very few plants that answer to this description. In nearly every industrial process in which steam is used for heating, drying or boiling, it is more economical to use a higher pressure in order to obtain the greater rapidity of result from the higher temperature. Even in paper mills and the finishing departments of textile mills it is usual to find pressures of from five to ten pounds used in drying In laundries, in breweries, in confectionery boiling, in wool and cotton drying and many other lines of work, 60, 80 and even 100 pounds pressure are usually employed, and in all these cases the pump and receiver method is wasteful. It must be remembered that in a dry room, for instance, on which 80 pounds pressure is carried, the water of condensation as it leaves the outlet, and while still under pressure, has almost the same temperature as the steam from which it originates, that is to say about 320 degrees Fahrenheit. When this water reaches atmospheric pressure it at once cools by expansion to the boiling point and throws off 108 degrees of its temperature and a considerable portion of its own bulk in the form of expansion vapor, some of which is a dead loss. It is manifest to the most inexperienced new hand that if this water can be carried back direct to the boilers without being allowed to expand there will be an absolute saving of the amount of coal necessary to reheat this water by the 108 degrees that it has lost. When one recalls the fact that steam or hot water under even five pounds pressure has a temperature 16 degrees; at 10 pounds, 28 degrees; at 30 pounds, 62 degrees, and at 60 pounds, 95 degrees above boiling point at atmospheric pressure, and that between 10 and 12 degrees increase in the temperature of the feed water means a theoretical saving of 1 per cent. in the fuel account, it follows as a natural consequence that it is wrong in theory as well as in practice to cool off water and then spend good money in replacing a loss of heat that can readily be retained and used over again.

If the water can be returned into the boilers above the evaporation point, at atmospheric pressure, the gain is very much larger in proportion than if the boiler feed is below the boiling point. The reason for this is that the working capacity of the boiler has been changed, and has been brought to a state where the boiler is more efficient in heat units. This difference of efficiency must be also credited to this method of feeding, and the change will naturally show greater total results than if the saving had been computed from changes due to the difference in the temperature of the feed water alone. I will give one interesting case out of a number I have met in my own experience.

In the Stormont mill of the Canadian Colored Cotton Mills Company, at Cornwall, Ontario, when the average temperature of the boiler feed was 165 degrees, five 80-horsepower boilers were used to do the drying and dyeing. Under the rearrangement now in force about 80 per cent. of the feed water goes back direct to the boilers through return traps at 280 degrees, the make-up water at 140 degrees, making the average temperature of the boiler feed 252 degrees Fahrenheit. This shows a gain of 87 degrees with a theoretical saving of about 8 to 9 per cent. in fuel. In reality, two of the boilers have been shut down, the other three are more efficiently operated, and the actual saving in fuel is over 30 per cent., allowing for the fact that a portion of the dyeing plant has been removed to another mill. This was originally a receiver and pump layout with the make-up water running into the receiver tank to cool the returns, and nothing was done by

*Abstract of paper read before the Canadian Society of Stationary Engineers, at Berlin, Can., July 27, 1910.

me except to handle the returns independently from the make-up. The cost of making the change actually pays for itself in this plant every six months.

In many sections of Canada, and particularly throughout central and western Ontario, another consideration is that of the quality of the feed water. Every engineer realizes the value of using the condensation over again, and how much it improves his general conditions to minimize the amount of fresh water that is fed into the boilers, and most of which in many places has to be paid for. And yet, for some unknown reason, I find many plants where the engineer appears to have failed to convince his owners. Only last week I was in a factory in western Ontario where the feed water is so bad that the 2-inch feed pipes to the boilers are practically choked up completely every four weeks by the heavy deposit of solids from the water. In this same plant, the exhaust from two engines and four pumps is utilized for heating water for manufacturing purposes, but every drop of this valuable condensation is allowed to run into the sewers. Enough distilled water is wasted to feed one 125-horsepower boiler practically continuously.

If it were a matter of enormous expense or engineering difficulty to save and use this water, one could understand the objection. But it is such a simple problem, and as this water is always at or near the boiling point, the saving in fuel would also pay for the cost in a reasonable time. All that is necessary is an oil-separation system, and a trap or pump that will lift this water back to the feed system.

An important matter is that of the proper trapping of steam-using machinery of every description. It is a point that has been very much neglected by the engineering profession. The steam trap in general has been condemned by many engineers and is a cause of dissatisfaction in many factories and plants. An examination of the conditions will often show the reason. The chief reason, in my experience, is that steam traps are seldom purchased intelligently. This may appear to be a strong statement to make, but it is an absolutely true stricture, and one which I am able to substantiate.

The first great cause for this is that extremely few engineers take the trouble to find out what the actual quantity is in pounds or gallons of water that is condensed in any given process or machine. The ratio of water condensed from steam varies enormously according to the service performed by that steam. It is ordinarily calculated that one square foot of radiation used for heating an ordinary building, will condense 0.625 of a pound of water per hour, when the building has been heated to 68 degrees Fahrenheit. But this will vary according to the construction of the building, the amount of glass surface and the humidity of the air, as well as its changes in a given time due to wind. Condensation in a dry room will vary according to the moisture of the goods being dried. A mangle in a laundry will condense more steam, or less, accordingly as the quantity of the goods passing over it in a given time is greater or smaller. Indirect coils where air is forced over the pipes will show enormous differences when the air is drawn from out of doors and when the air is recirculated within the building. A vacuum pan will condense steam much more rapidly than a pan in which the same liquid is boiled at atmospheric pressure. Every process shows different results, and a proper test of the quantities almost invariably shows that the ordinary methods of guesswork run from 25 to 50 per cent. too low. It is quite usual to find a steam trap with an interior valve opening of ¼ inch, and actually able to discharge about 10 pounds of water per minute with five pounds pressure, placed where it has to take care of double that quantity of water, a thing that it is not possible to do. The trap floods, it refuses to do work that is against all natural laws, and in consequence the trap is blamed and not the man who attempted to force it to do impossibilities. A certain amount of blame also attaches to the engineer, because the chances are that he probably did not by actual personal investigation find out for himself the size of the opening through which the water had to pass. On the other hand, a certain amount of blame attaches to the makers of these traps, who fail to inform the engineering public of this most important detail in trap construction.

Ordinary so called catalog ratings are useless. They are usually based upon a fixed pressure of, say 50 pounds, and their drainage capacity in feet of 1-inch pipe is misleading. No allowance is made for varying conditions, and a trap that will handle a given quantity of water at 50 pounds will do nothing like that amount at 5 or 10 pounds pressure.

Every engineer knows that in order to obtain from steam the heat units which he wishes to utilize, he must first condense that steam. Every engineer knows that in order to obtain the best results, the apparatus must be drained quickly, and most engineers know that condensation does not usually leave the apparatus in a steady stream, but that the water nearly always comes in gulps and sudden gushes. It is this maximum condition that must be taken care of by the trap, and the most economical and useful trap is that which is a little larger than the actual average condition calls for.

With steam traps, as with every other kind of machinery, it is mistaken economy in every way to buy too small-sized units. The overloaded boiler and the overloaded engine are expected to give poor results in the long run, and yet I think I am not in the least guilty of exaggeration when I make the statement that fully 50 per cent. of all the steam traps in use in this country are overloaded and expected to do work of which they are not capable. This means another direct loss in the efficiency of the apparatus being drained, and in many cases reduces the output as much as one-half.

I wish to emphasize this important feature, and it covers the two great points to be aimed at in the economical handling of condensation, namely, that, first, all steam-using machinery should be amply and quickly drained, and, secondly, the water of condensation kept under pressure and returned direct to the boilers without loss of its heat units.

The method of doing this is simple. A proper system of traps will do the work automatically. The upkeep expense is practically none, and depreciation allowance less than that of any other part of an ordinary power plant. Return lines should be covered to retain the heat in the water; this is a practice which I always suggest and which the results justify.

It is usual in most steam-using plants to cover the supply pipe and leave the return pipes uncovered. You will tell me that the reason for covering the supply lines is to prevent excessive condensation in the line through radiation of heat through the pipe. My argument goes farther. Prevent that same radiation loss in the return. It represents just as much coal on the return side as on the supply side. If you can force the water into the boilers under pressure as hot as you get it, why allow any unnecessary loss at either end? Pipe covering only represents a small first cost; loss of heat by radiation represents a perpetual expense in coal.

By a proper trap I mean one of ample discharge capacity, and not a trap of the kind that requires a cooling of water to open its valve, such as the usual expansion type. Remember that the object is to utilize as fully as possible every heat unit contained in the water as it leaves the place of condensation. The rest is merely an application of ordinary common sense and reasoning power. Get the condensation back above the boiling point at atmospheric pressure, and you are making money.

It is difficult to imagine what argument can be set up against the economy of this method, and yet I find that many owners and engineers of plants where the old methods exist have the greatest prejudice against venturing on a change that is such a simple and reasonable one. I have made these changes in a great many plants, and in every case the owners have expressed their regret that the change had not been made sooner.

Electrical Department

Especially conducted to be of interest and service to the men in charge of the electrical equipment.

Storage Batteries in Isolated Plants

By Norman G. Meade

A system of battery-voltage regulation for use in installations of comparatively large size introduces the use of a motor-driven shunt-wound booster connected so that its voltage is added to that of the generator, "boosting" the voltage to the amount required for charging the battery. The charging current is controlled by the booster field rheostat and the voltage on discharge is controlled by means of end cells.

The booster is simply an ordinary generator having its field winding connected across the station busbars. The armature winding is connected in series with the battery and its function is to add to the line voltage sufficient "boosting" voltage to force current through the battery. If, for example, a 110-volt generator supplies a 110-volt circuit to which the battery discharges, 60 battery cells will be required, as explained. To bring these cells up to 2.65 volts per cell on charge requires a potential of 159 volts. The booster must furnish the difference between the maximum voltage and that of the generator, for this case 159 — 110 = 49 volts. The armature capacity must be sufficient to carry the maximum charging current for two hours without overheating. These factors fix the size of the machine. Usually the booster is motor driven. The kilowatt capacity of the motor can be less than that of the booster, because the booster never delivers its maximum current except at a low voltage and the overcharge current is usually half the normal eight-hour rate. If the kilowatt capacity of the motor be from 65 to 75 per cent. of the booster rating it will generally be amply powerful for its duty.

Fig. 7 shows an elementary diagram of a battery, with a charging booster, arranged to discharge through end cells. With the connections as shown the booster adds its voltage to that of the generator and the combined electromotive force is sufficient to send current through the battery. When the battery is to be discharged, the single-pole double-throw switch is thrown over so that the blade is in the upper jaw, connecting the battery directly across the line.

The rate of charge is controlled by the booster-generator field rheostat which controls the booster voltage. This rheostat should have a sufficient number of steps and a high enough resistance to vary the booster voltage from a minimum of two volts to the maximum, by steps not exceeding three volts each. The field winding is always connected to an outside source of energy, never to the booster brushes. The connections of a switchboard adapted to this system is shown in Fig. 8. Three circuit-breakers are shown in the diagram, two overload breakers, one for the battery and one for the booster motor and one underload breaker to prevent reversal of current through the generator.

In selecting a system to be used attention should be directed to the following points: The equipment for charging in parallel and discharging in series, with rheostatic control, is the least expensive to install and is the one generally used where low first cost, simplicity of construction and ease of operation are of more importance than economy. The system using end cells and switches is adapted to plants of the same size as the first system, but is usually more expensive to install, except where the size of the generator is fixed by the charging rate of the battery, as is often the case. The battery being charged in series requires a generator of 155 volts, but the

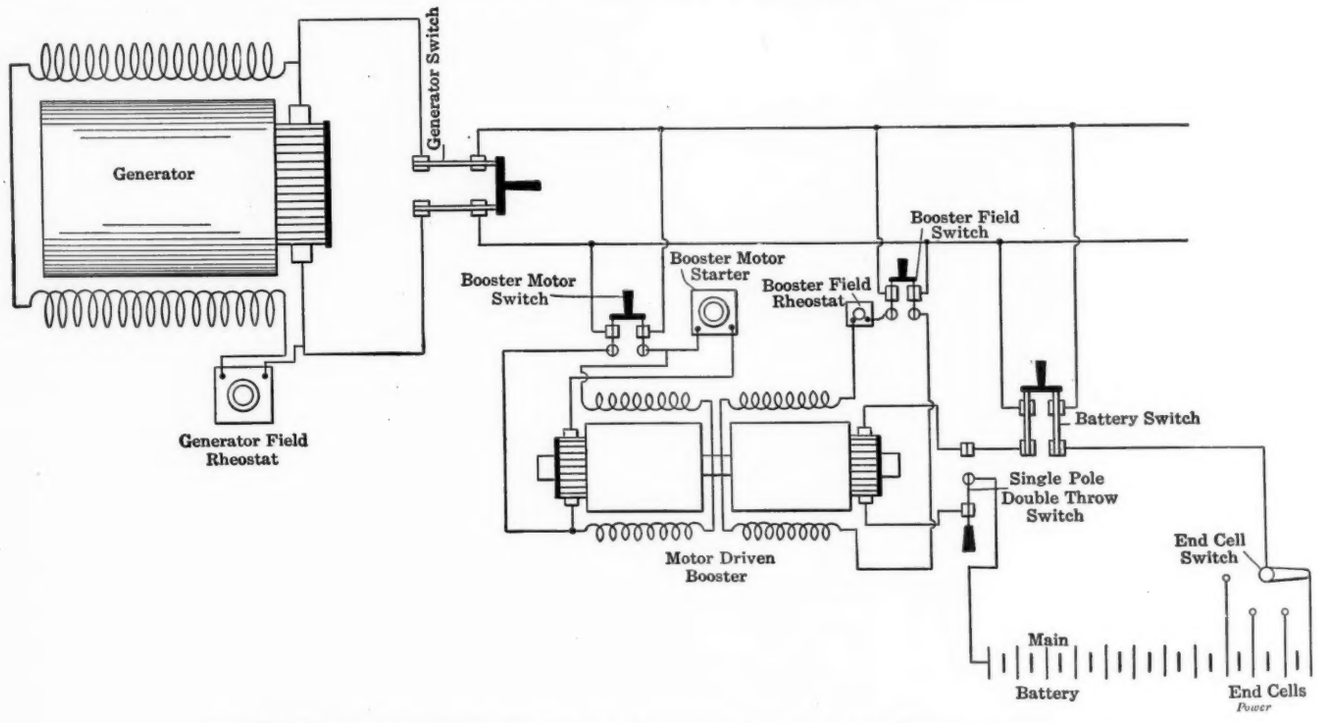


Fig. 7. Elementary Connections for Combined End-cell and Booster Regulation

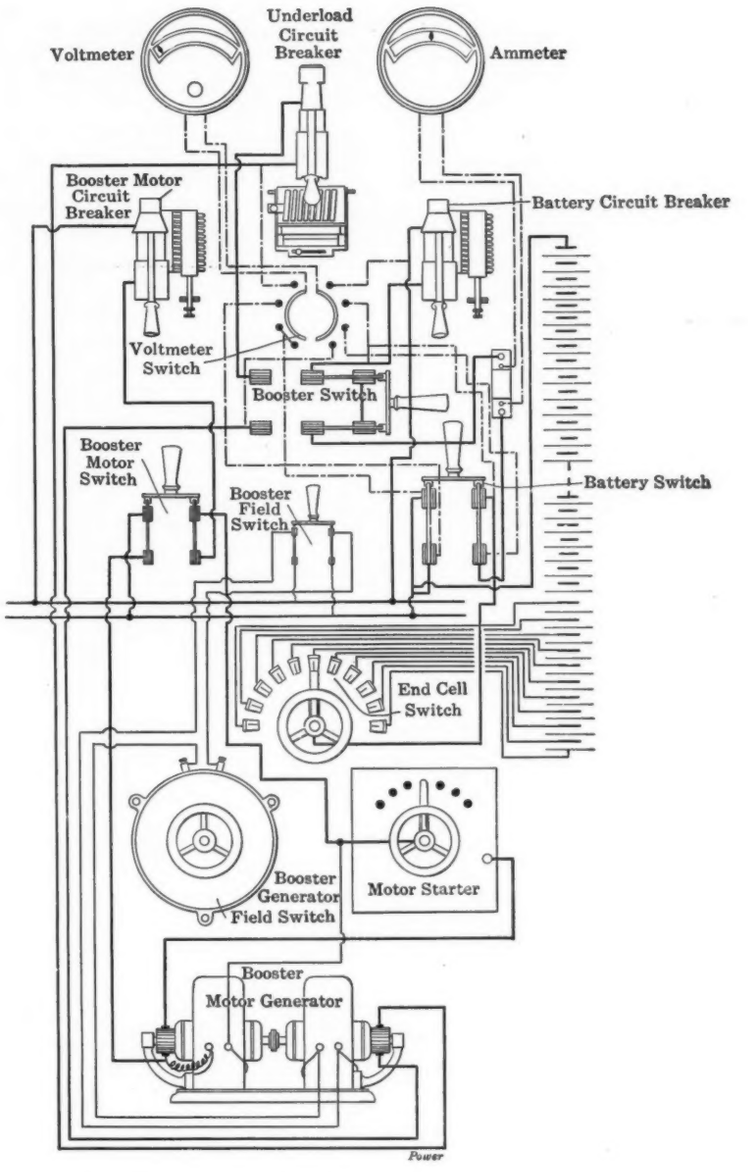


FIG. 8. SWITCHBOARD FOR COMBINED END-CELL AND BOOSTER REGULATION

generator ampere capacity required is one-half that required for the first system. The additional cost due to the end-cell switches must be taken into consideration when comparing the two systems.

The motor-driven booster system is the most satisfactory for plants of comparatively large size, where the generator has a current capacity considerably larger than the normal charging rate of the battery.

In selecting the size of battery to give a certain discharge, care must also be taken that the generator is large enough to charge the battery at a rate not lower than the normal eight-hour rate. In the first system described, where the two halves of the battery are charged in parallel, each half taking the normal rate, the generator must have a current capacity twice that of each half of the battery.

It is well to have the generator capacity sufficient to charge the battery occasionally at a rate somewhat in excess of the normal, as this not only keeps the cells in better condition but makes it possible to shorten the length of the charging period when conditions warrant this. With the first system, a high rate of charge can be obtained without excess generator capacity by charging each half of the battery separately, but the charging time under these conditions would be nearly twice as long as usual.

In isolated plants supplying a widely fluctuating power load, such as elevators, a nonreversible automatic booster can be used to advantage in connection with storage batteries. It causes the battery to charge and discharge as the external load increases or decreases. Fig. 9 shows the manner in which the booster is connected in circuit. The lighting load is connected across the generator mains as shown and the motor load to the mains beyond the booster and battery. The booster is excited by a compound field winding, in which the series and shunt

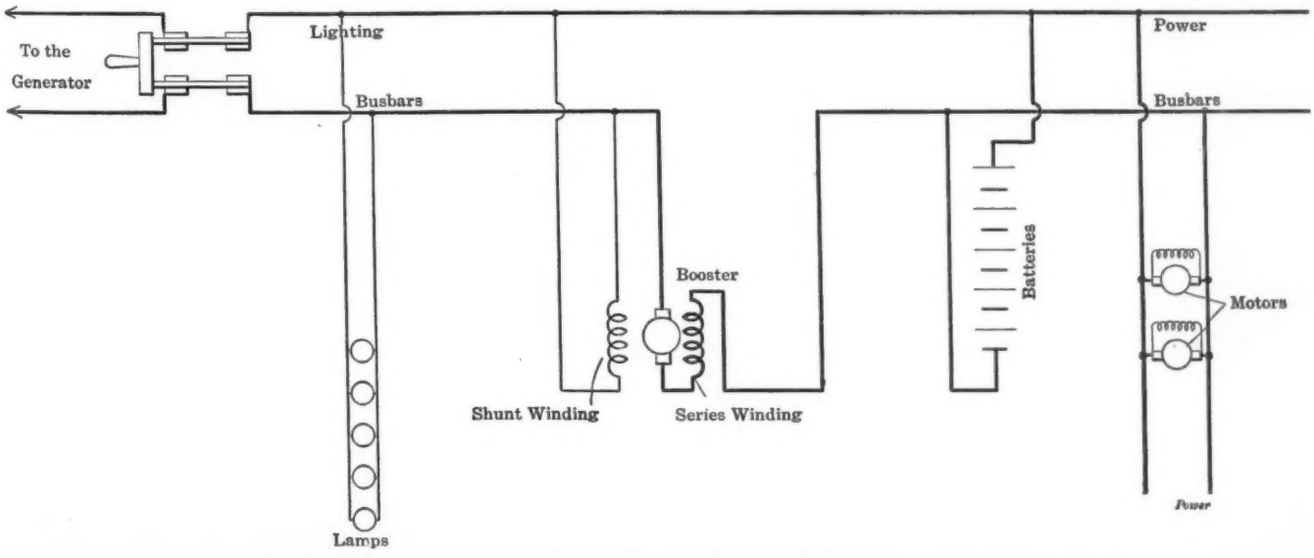


FIG. 9. ELEMENTARY CONNECTIONS FOR A "FLOATING" BATTERY AND BOOSTER

windings are connected in opposition, the series winding tending to produce a booster voltage opposing that of the generator while the shunt winding tends to produce an electromotive force assisting that of the generator. The shunt winding is stronger than the series winding; therefore the booster voltage is in the same direction as that of the generator. The voltage at the power busbars is therefore greater than that at the lighting busbars, the difference being about 15 volts, which is the usual normal electromotive force of the booster.

The battery electromotive force, when there is neither charge nor discharge current passing, is equal to the normal electromotive force of the mains across which it is connected. When the current used by the motor load is equal to the normal booster current, all current through the booster armature and series field winding goes to the load and the battery neither receives charge nor discharges. If the motor load decreases, the current through the booster armature and series winding will decrease, thereby reducing the magnetization of the latter, which strengthens the resultant booster field. This increases the voltage between the motor load busbars, thereby sending current to the battery. By properly proportioning the field windings, the current through the booster is reduced very slightly, the excess of current over that required for the load charging the battery. Conversely if the load should increase, a small increase in current through the booster increases the excitation of its series winding, which reduces the booster electromotive force. The voltage between the power busbars is thereby reduced and the battery therefore discharges, adding sufficient current to the booster circuit to make up for the amount required for the increase in load.

Speed Classification of Motors

The wide range of motor applications and the high degree of adaptability to different service conditions which characterize the electric motor have led to the development of a number of classes of motor, on the basis of speed characteristics. Besides the "old reliable" shunt-wound machine which runs at almost constant speed with any load, there are motors which change their speeds as much as 20 to 40 per cent. between no load and full load; others which can be run at any one of a few widely different speeds, maintaining practically constant speed at all loads, and still others that can be run at any one of a large number of speeds, each differing slightly from the one next in rate.

In view of this development, the Westinghouse Electric and Manufacturing Company has wisely adopted a standard classification of motors according to their speed characteristics, as follows:

Constant-speed motors, which do not vary greatly between no load and full load. These include synchronous motors; ordinary induction motors; shunt-wound direct-current motors and compound-wound machines which do not run more than 20 per cent. faster at no load than at full load.

Multispeed motors, meaning those that can be operated at any of several distinct speeds, each practically constant. These include induction motors in which the primary windings can be grouped so as to form different numbers of magnetic poles; and direct-current shunt-wound motors with two armature windings. The horsepower varies with the speed.

Adjustable-speed motors. This class is restricted, so far, to shunt-wound and moderately compounded direct-current machines of which the speed can be adjusted gradually over a considerable range by varying the field strength. Each speed is practically constant regardless of load, and the maximum horsepower is practically the same at all speeds.

Varying-speed motors, in which the speed varies almost in inverse proportion to the load, such as series-wound machines and those with compound windings heavy enough to cause more than 20 per cent. increase in speed from full load to no load.

Motors provided with heavy series field windings which are used only at starting and are cut out at full speed do not come within the last class. They are classed as shunt-wound machines because at normal speed they are of that type.

LETTERS

An Erratic Power Factor

An underground cable about three miles long transmitted 60-cycle alternating current to incandescent lamps which were supplied from transformers originally built for 125 cycles; these received current at 2200 volts and delivered it to the lamps at 110 volts. At very light loads the power factor of the primary circuit (the three-mile cable) was 85 per cent., at a little more than half load it went up to 100 per cent. and at full load it dropped to 90 per cent. This variation in the power factor was, of course, unusual and seemingly erratic. My explanation of it is that during light loads the inductance of the transformers was rather high, so that the current lagged considerably behind the electromotive force, tending to give a low power factor. This was partly neutralized by the capacity of the cable, so that the power factor was not as low as would be expected from the lagging current. As the load came on the transformers, the capacity of the cable remaining constant, or practically so, the inductive effect of the transformers was a very much smaller proportion than under light load, so that at a little above half load the power factor was unity, the inductance just balancing the capacity. With a further increase in load, the inductive effect still diminishing in its proportion, the capacity effect overbalanced the inductive effect and the power factor decreased until at full load it was approximately 90, the current then leading the electromotive force. That this condition of affairs was the cause of the trouble was satisfactorily proved to my mind by the fact that a number of the other circuits which were air lines, but practically the same otherwise, showed power factors under light loads considerably less than the cable circuit, and never approached unity very closely at full load. These circuits, however, went through exactly the same cycle of changes as the first cable circuit when they were changed to cable.

I should be pleased to have other readers give their opinions of this peculiar case.

HENRY D. JACKSON.

Boston, Mass.

[Mr. Jackson's explanation is evidently the only one that can be considered, unless there were modifying conditions which escaped his notice. The power factor of a transformer is very low at no load and small loads, and increases as the load increases, approaching 99 per cent. at full load. This is due to the increasing magnetic reaction between the primary and secondary windings, as the secondary current increases. At no load there is no secondary current and the transformer core and primary winding constitute a highly inductive reactance coil.—EDITOR.]

A Simple Polarity Indicator

I have read in the columns of POWER descriptions of several ways of finding the polarity of a direct-current dynamo, all of which are good. But there is a very simple method that I have used to advantage which seems to be unknown to other writers on the subject. Take a common water tumbler nearly full of water, connect a small copper wire to each of the dynamo terminals or to the main switch blades, and dip the other ends in the water, moving them slowly toward each other until small air bubbles form on the end of one of the wires; this will be the negative wire. This method applied to 110- or 125-volt machines has never failed with me.

R. T. WHITE.

Waterbury, Vt.

The lowest atmospheric temperature ever observed, —69 degrees Centigrade (—90.4 degrees Fahrenheit), was recorded on January 15, 1885, at Werchojansk in Eastern Siberia.

Gas Power Department

Tar Extractor for Bituminous Coal Gas

The accompanying illustrations show an apparatus devised by Prof. F. W. Burstall, of Birmingham (England) University, for extracting or separating suspended matter from gases, particularly tar from combustible gases. Fig. 1 is a longitudinal section, and Fig. 2 a transverse section of the apparatus suitable for extracting tar from combustible gas. It comprises a box *A* provided with a removable lid *B*, gas inlet and outlet *C C*, and a water-sealed bottom *D*, it being otherwise entirely a gas-tight structure, maintained also in this respect by the lid, and the inlet and outlet branches. In connection with the lid *B* are provided a

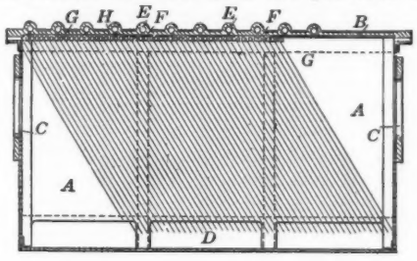


FIG. 1. TAR EXTRACTOR FOR GASES

number of water passages *E*, through which numerous small openings *F* communicate with the interior of the box in such manner that the water is jetted or sprayed into the interior of the apparatus.

A number of wires extend at small distances apart diagonally across the interior of the box, with their one ends *G* fixed to carrying bars *H* and their other ends free, and as all these wire lengths are between the inlet and the outlet, and

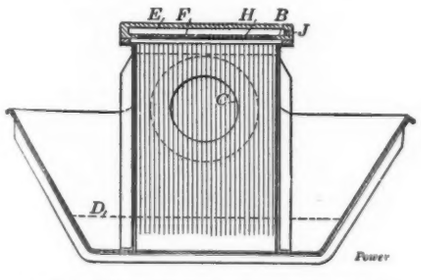


FIG. 2. TAR EXTRACTOR FOR GASES

extend almost the full depth of the interior of the box, the gas passing through is broken up and made to pass in quite a circuitous manner between the wires. Each of the wires is able to vibrate independently, and it is not intended that they should touch at any time. The water inlets are arranged at an angle to the wires.

In action it is believed that the wires vibrate within a dense water vapor or atmosphere provided by the liquid, and that the tar in suspension clings to the wires and gradually trickles downwardly off their free ends into the bottom of the box, whence it can be removed, the vibrating or quivering of the wires being effected by the combined actions of the gas passing through the box and the water spraying on to the wires. The water seal at the bottom of the box is sufficient to prevent any gas finding its way therethrough, and it is intended that the same water may be used over and over again by pumping the surplus of the seal into a pressure tank or pump supply connected up to the passages *E*, which passages have their inlets *J* alternately placed on opposite sides.

In addition to the extraction of tar, the gases treated may be cooled by the action of the apparatus.—*The Mechanical Engineer* (London).

Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men.

Elementary Lectures on the Gas Producer*

BY CECIL P. POOLE

Before going any farther in the study of what goes on in a gas generator, it is advisable for the student to get a general knowledge of the different fuels used in producers and a little more information about gases and their behavior under the influence of heat.

SOLID FUELS

As the reader probably knows, there are many grades of solid fuel, not counting wood. The first stage in the formation of coal is the decomposition of vegetable matter in marshes. This produces what is known as "peat." The next stage of evolution produces "lignite," which is a soft coal with a moderate proportion of carbon and a large proportion of moisture. Next comes bituminous coal, with still more carbon and less moisture; finally, there is anthracite, which is mostly carbon. Between "straight" bituminous and "straight" anthracite, however, there are several grades, and the difference is so great that these have been classified into semi-bituminous and semi-anthracite—highly inappropriate names, though fairly descriptive when one allows sufficient latitude in defining "semi."* Semi-anthracite is "near-anthracite" and semi-bituminous is "near-bituminous" coal.

All coals contain solid carbon and several gases, most of the latter being combustible. In addition to the carbon and gases, coal contains sulphur, moisture and a number of noncombustible solid substances, such as silica, aluminum and iron, which are lumped under the head of "ash." The gases contained in coal are hydrogen, oxygen, nitrogen and compound gases called "hydrocarbons" because they are composed of hydrogen and carbon.

Carbon is the most valuable constituent in any producer fuel, because when it is completely burned the product of combustion (carbon dioxide) does not carry away any heat energy with it, except the small amount used to raise its temperature. When hydrogen is burned, on the other hand, the hydrogen and oxygen combine to form water, vapor or steam, and this carries away with it the latent heat of evaporation in addition to the heat used in raising the temperature of the steam above 212 degrees. These questions will be discussed more fully in a future lecture on heat and combustion.

There are two kinds of analysis to which coals are commonly subjected; one is called "proximate" analysis and the other is called "ultimate" analysis. The proximate analysis gives the proportions of fixed carbon, volatile matter, moisture and ash. "Fixed" carbon is carbon not chemically combined with anything else, and "volatile matter" includes all gases except those parts of the hydrogen and oxygen which constitute moisture.

The ultimate analysis gives the proportions of carbon (including both the fixed carbon and that combined with other elements), hydrogen, nitrogen, sulphur and ash. There also remains some oxygen, a small portion of which is "free," the most of it forming eight-ninths of the moisture. It is impossible to determine directly the proportion of oxygen in coal,

*This is the fourth lecture of the series.

*"Semi" means one-half, but is frequently used to mean "partly." Even with this definition, "semi-anthracite" does not really mean what it is intended to mean.

however, so the custom is to determine the proportions of the five constituents just mentioned and assume that what is left is oxygen. The proportions are determined by weight; that is, if a coal contains 75 per cent. carbon, each pound of coal contains three-fourths pound of carbon, and so on down the list of constituents.

Table 1 gives some representative analyses of four kinds of coal and one of lignite. Referring to the proximate analyses, it will be found that the percentage of fixed carbon divided by the percentage of volatile matter is larger and larger, reading from left to right in the table. That is, F. C. ÷ V. M. for the Montana lignite is about 1.19, while the ratio for the bituminous coal is 2.15, that for the semi-bituminous is 3.58, that for the semi-anthracite is 10.2 and the ratio for anthracite is 14.9. This ratio of fixed carbon to volatile matter is generally considered the factor which determines the class in which a coal belongs. There is no sharp dividing line between the adjoining classes that is recognized by legal or other authority. It is customary, however, to call all coals bituminous in which the proportion of fixed carbon is less than three times that of volatile matter. Coal in which the fixed carbon is 12 times the volatile matter, or more, is classed as anthracite. The dividing line between semi-bituminous and semi-anthracite has never been agreed upon.

TABLE 1. DIFFERENT ANALYSES OF COAL.

Kind of Analysis.	Constituents.	Montana Lignite.	West Virginia. Bituminous.	West Virginia. Semi-Bituminous.	Pennsylvania. Semi-Anthracite.	Pennsylvania. Anthracite.
Proximate	Fixed carbon	42.08	61.55	71.25	71.79	84.95
	Volatile matter	35.90	28.58	19.88	7.02	5.70
	Moisture	11.05	1.48	3.94	5.41	3.45
	Ash	10.97	8.39	4.93	15.78	5.90
Ultimate	Total carbon	59.08	77.82	79.78	72.65	86.45
	Total hydrogen	5.37	4.89	4.60	3.10	2.38
	Nitrogen	1.33	1.48	1.01	0.77	0.75
	Sulphur	1.73	0.90	1.16	0.74	0.29
	Ash	10.97	8.39	4.93	15.78	5.90
	Oxygen, etc	21.52	6.52	8.52	6.96	4.22
Derived	Total carbon	59.08	77.82	79.78	72.65	86.46
	Available hydrogen	4.14	4.73	4.16	2.50	1.62
	Sulphur	1.73	0.90	1.16	0.74	0.29
	Nitrogen	1.33	1.48	1.01	0.77	0.75
	Moisture	11.05	1.48	3.94	5.41	3.45
	Ash	10.97	8.39	4.93	15.78	5.90
	Unknown	11.70	5.20	5.02	2.15	1.53
Heat value; B.t.u. per pound = ...		10,539	13,718	14,382	12.047	13,470

Referring again to Table 1, it will be found that in addition to the proximate and ultimate analyses there is given a "derived" analysis; this is an analysis derived from the other two analyses, not made directly from the coal. This analysis differs from the ultimate analysis in three points: It gives the proportion of useful hydrogen instead of total hydrogen, the proportion of moisture and the proportion of actually unknown constituents instead of lumping them with oxygen.

The useful hydrogen is that which is not contained in the moisture; the latter is of no use as fuel, because just as much heat is used up in decomposing the moisture into hydrogen and oxygen as is liberated or made available by burning the hydrogen. Moreover, when the hydrogen is burned, the resulting steam is superheated far above the temperature of evaporation (212 degrees at atmospheric pressure), and this absorbs still more heat. The proportions of carbon and useful hydrogen, therefore, are much better indications of the true fuel value of the coal than either the proportions of carbon and total hydrogen or those of fixed carbon and volatile matter.

Take, for example, the analyses of the Montana lignite. The proximate analysis shows 11.05 per cent. of moisture, or water, and the ultimate analysis shows 5.37 per cent. of hydrogen. Of the moisture, one-ninth is necessarily hydrogen, and this one-ninth is

$$11.05 \div 9 = 1.23$$

per cent. of the weight of the coal. Subtracting this from the percentage of total hydrogen gives

$$5.37 - 1.23 = 4.14$$

per cent. of hydrogen that is available as fuel. The total proportion of actual fuel material, therefore, is 59.08 carbon + 4.14 hydrogen + 1.73 sulphur = 64.95 per cent. The proximate analysis, however, shows 77.98 per cent. and the ultimate analysis 65.78 per cent. of combustible material, the former being very much, and the latter slightly, greater than the actual combustible material, if the analyses were accurately made.

The derived analysis is clearer than the others also in showing the true proportion of unknown substances in the coal; in the proximate analysis these are included in the volatile matter and in the ultimate analysis they are included with the oxygen.

Upon comparing the analyses it will be evident that the "volatile matter" content commonly given in proximate analyses includes sulphur. This does not seem a rational plan, because the sulphur that is commonly known is not volatile. But the sulphur contained in coal *is* mostly in volatile form, supposedly as sulphide of iron. However, since the exact character of the sulphur in coal is not positively known, it is more nearly accurate to state the sulphur content separately, although it is so small that it is not of much importance, and it is practically impossible to analyze coal with exactness.

Coke and Charcoal

Besides the coals just discussed, there are two solid fuels known as "charcoal" and "coke." These are not strictly additional fuels, because they are made from wood and coal by distilling out the volatile constituents of those fuels and leaving the solid carbon and the ash (mineral impurities). Charcoal is made from wood and coke is made from bituminous coal. The process consists, broadly, of heating the wood or coal moderately—not hot enough to set it afire—and maintaining the heat until practically all of the gaseous contents are driven out of it. The temperature at which the wood is charred is somewhere between 400 and 600 degrees Fahrenheit. The temperature to which coal is heated to drive off its volatiles is around 1600 degrees Fahrenheit. The higher the temperature of charring or coking, the less charcoal or coke will be obtained from the wood or coal, within certain limits. Slow distillation (moderate temperature) produces the largest percentage of charcoal or coke.

The weight of charcoal obtained from a pound of wood varies from one-fifth to one-third pound ordinarily, and the weight of coke obtained from coal ranges from about one-third to seven-eighths of the original weight of coal.

Since the ash is all left in the charcoal and coke, and the weight of material is very much reduced, the percentage of ash in charcoal and coke is very high, compared with that in the original fuel. Charcoal and coke, when first made, contain no moisture, but both of them absorb moisture greedily. Consequently, when they reach the market they are several per cent. heavier than when first taken from the charcoal pile or the coke oven, because of the moisture absorbed in the meantime.

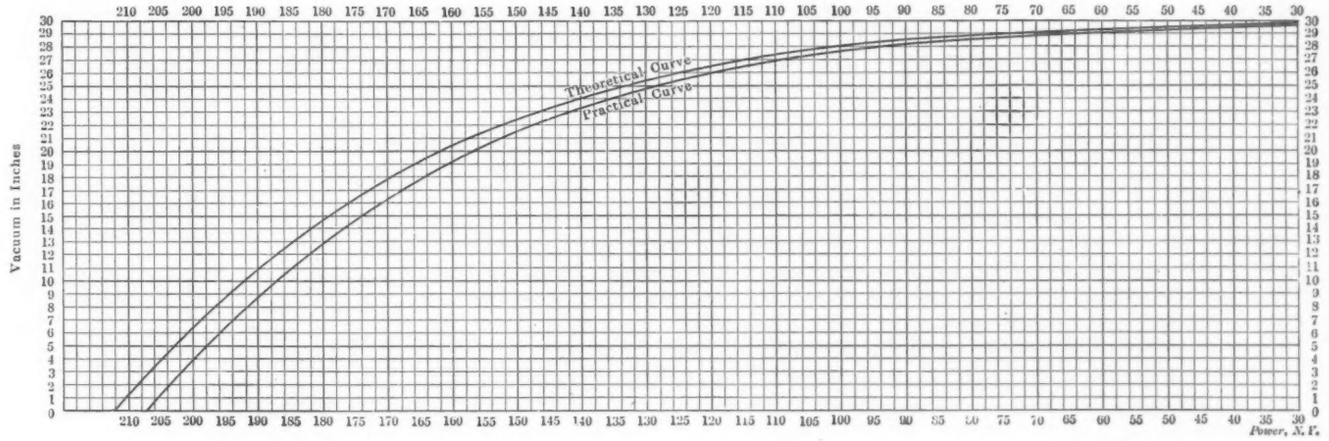
Neither charcoal nor coke is used as a fuel for gas producers, for the reason that they are much more expensive than the fuels from which they are made, and these fuels can be used in producers. For this reason it is not worth while to discuss the composition and fuel characteristics of charcoal and coke in this series of lectures. The general discussion of solid fuels, however, would be incomplete without any reference to charcoal and coke.

Readers with Something to Say

Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words, wanted.

Condenser Economy

In the economical operation of condensers it is very important for the engineer to know the proper temperatures of the injection and discharge waters which will give him the best vacuum and the most economical results. The accompanying chart will prove a valuable aid in the economical running of a condenser. The temperature of the injection water is usually beyond the control of the engineer and varies from summer to winter temperatures, making it necessary for him to supply more or less water to the condenser to maintain the same vacuum. The temperature of the discharge water must therefore be the guide of the engineer in determining whether he is getting all the vacuum possible, or whether he is pumping too much or too little injection water.



TEMPERATURE, FAHRENHEIT, OF DISCHARGE WATER FROM CONDENSER

By referring to the chart it will be seen that with 120-degree discharge water, 26½ inches is the greatest vacuum attainable. With 100-degree discharge water 27¾ inches of vacuum is attainable. The relation between the discharge temperature and the attainable vacuum is due to the boiling point of water changing under the various vacua.

If your condenser shows a discharge water temperature of 120 degrees or less and you are not obtaining close to the vacuum attainable as shown on the chart, then you have leaks or uneconomical conditions that should be investigated.

Again, if 25 inches of vacuum is all that is attainable with your plant, due to construction features, and your discharge water shows a temperature of less than 128 degrees, then it would be possible to use less injection water and still maintain the same vacuum, thus effecting a saving proportional to the decrease in speed of the pumps and in the amount of steam thus saved.

This chart, kept in an accessible place and referred to occasionally, will tend to increase the efficiency of your plant.

N. L. DANEY.

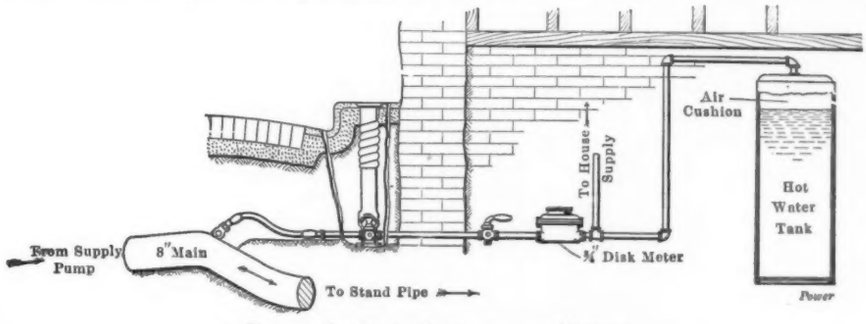
Elyria, Ohio.

A Creeping Water Meter

It is not uncommon for an electric meter to "creep" with the customer actually consuming no power, but for a water meter to perform this singular feature is almost unheard of. However, here is the case of a water meter correctly geared and registering 40 gallons per hour without a gallon being used or wasted.

This meter was of the disk type and had been in service for over two years, never having registered over 2000 gallons per month. The discrepancy was at once noticed by the auditing department and a man was sent to check the reading. This he did, then read it again in two days and found it to be averaging about 1000 gallons per day. He also reported that the meter was observed to be running. The natural conclusion was a leak, so the bill was sent in. The customer immediately made a strenuous complaint, stating that he had not used more water than during any month previous.

The piping was carefully gone over and no leak being discovered the conclusion was reached that the waste had been stopped. On going back to the meter, however, it was found to be still running. On closing the customer's cutoff valve the meter stopped and on opening it immediately started up. The meter was taken out and tested but was found to be correct. The next idea, however, solved the problem—that of inserting a check valve on the pipe line between the



PIPING SHOWING LOCATION OF METER

cutoff valve and the meter. The explanation of the trouble is as follows:

The meter was connected as shown in the accompanying sketch. The main, passing over a hill, was bent as exaggerated in the drawing and the tap was made at about the highest point of the bend. A four-cylinder pump geared direct to an induction motor pumped water to the standpipe through this main. One of the pistons was put out of commission about a month before and this probably produced a surging in the main from the standpipe to the pump. An air cushion was thus formed at the top of the hot-water tank. This surging compressed and released this air cushion at every part of the revolution of the pump in which this broken piston passed and caused the water to pass back and forth through the meter. The disk-type meter will not run backward although it will admit water through it in the opposite direction. This, of course, caused the meter to register constantly even though no water was consumed.

A. E. HOLMAN.

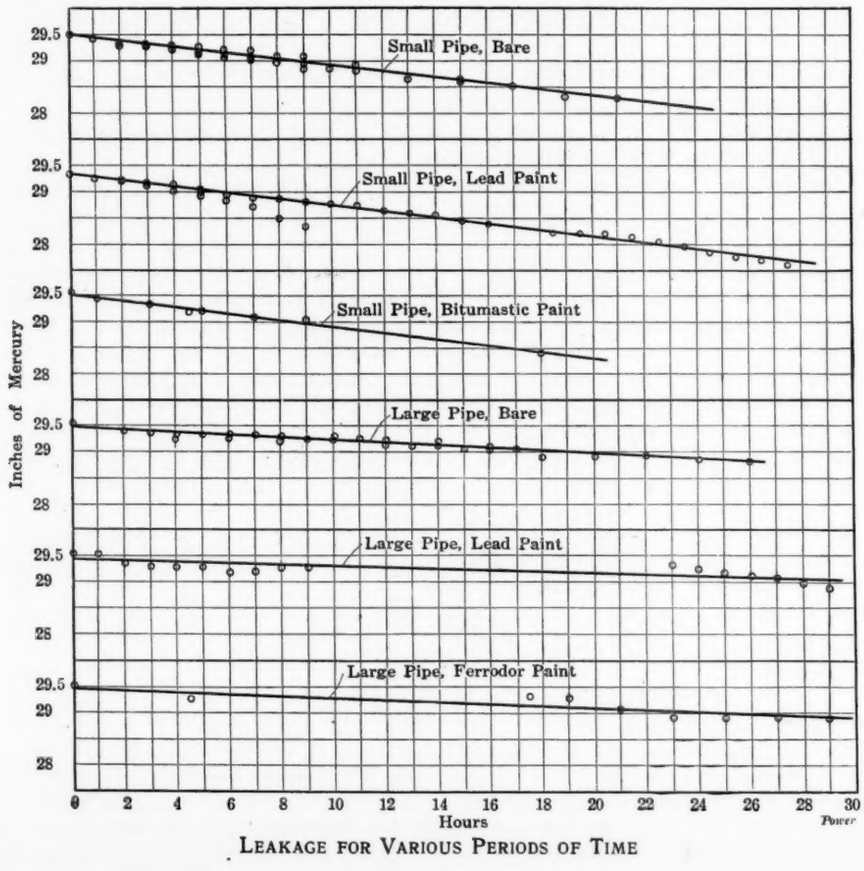
Anderson, S. C.

Air Leakage through Cast Iron

Some time ago the writer made a few tests on some cast-iron pipe to determine if air leaked through cast iron directly; that is, through the grain of the iron itself. The tests were simple ones, and the results obtained are of a comparative nature.

Two pipes were used in the experiment; the one, designated as the small *pipe* on the curves, was about 9 inches inside diameter by 6 feet long and the other, referred to as the *large pipe,* was about 14 inches inside diameter and 6 feet in length. The pipes were covered at the ends with blank flanges fitted with ⅛-in. sheet packing. A mercury column was connected to one end of the pipe, the other end being connected to a three-throw high-vacuum Packard pump. The tests were carried on under a high vacuum, and only the rate of leakage was noted in the results, as no attempt was made to measure the quantity of air.

The experiment was performed in the following manner: The vacuum being put on the pipes, all connections to the pump were closed and readings were taken, first at hourly intervals, and then at longer periods, since the drop in vacuum was very small. These readings were corrected by the barometer; that is, the difference between the actual barometer reading and 30 inches was added to the reading on the mercury column at the end of the pipe. These corrected readings were plotted against time, and the slope of the curves represents the rate of leakage. All efforts were made to prevent any leakage at the joints at the ends by keeping these parts well covered with a heavy paint, but still an uncertainty here existed. First, the pipes were tried bare, then they were covered with a lead paint. This paint was removed with lye and hot water, and a special paint known as Bitumastic paint was used on the small pipe, and Ferrodor paint on the large pipe.



LEAKAGE FOR VARIOUS PERIODS OF TIME

It can be seen from the curves that a very small amount of drop was experienced in the tests, some of them running 30 hours. Also for the same pipe the curves are nearly all parallel, which indicates that the rate of leakage through all the different paints and on bare pipe was the same. This tends to show that no air actually leaked through the cast iron, but this statement may be taken as of comparative value only. However, it seems that the greater leakage in pipe lines occurs through the joints and around the prime mover. Compared to the above leakage, the leakage through the pipe itself is probably negligible.

The experiment also shows that joints may be made tight if sufficient care is taken, as each pipe had two joints through which leakage might occur.

G. A. GLICK.

Joliet, Ill.

Faulty Design and Workmanship

Here in New England, many boilers have been installed with an outside manhole fitted as shown in Fig. 1, having a flange with a raised surface *C*, 1¼ inches wide, extending around the main opening. This is evidently done to save machine work, and the usual method of packing is to use an ordinary molded gasket about 3/16 inch thick. When the bolts are screwed up fairly tight, the flanges outside of the bolt circle are about 5/16 inch apart and further tightening of the bolts is apt to place excessive stress on the flanges.

One of our boilers was fitted in this manner and had been out of service for washing and general fixing up. Before leaving in the evening I left word to cut out one of the other boilers and cut in the clean one after the peak load had passed. Returning to the station next morning I found a slow fire under the boiler and steam just beginning to show. The fireman said he had tightened the bolts on the manhole cover but was afraid that something was wrong with it, so I immediately went on top of the boiler and found the main casting broken as shown at *A*.

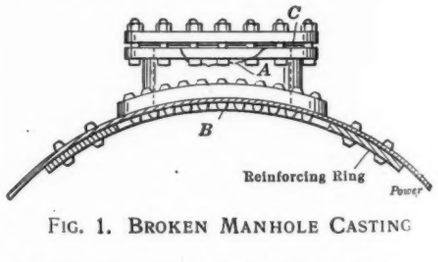


FIG. 1. BROKEN MANHOLE CASTING

Upon taking off the cover and striking the edge of the flange with a hammer the entire flange was easily knocked off and in a number of places there was

evidence that the grain of the iron had been disturbed to a depth of ½ inch.

The manhole was 12x16 inches, or equal to an area of 150 square inches, which at 125 pounds gage pressure equals 18,750 pounds tending to force the manhole cover off. The shape of the casting is such that the tail *B* of the rivets had to be on the outside, they being driven from the inside before the tubes or horizontal stays were put in.

FIG. 2. INSIDE MANHOLE COVER

The photograph of the four rivets marked *X* shows a fair average of the thickness of the rivet heads as compared with that of a standard rivet. Twenty-nine of these held the casting to the boiler shell. The thin rivet heads were inside the boiler and, although much out of proportion to the body of the ⅞-inch rivets, were allowed to pass, but would not have been permissible on the outside of the shell. This proves the truth of the statement in a recent issue of POWER: "The careful inspection of steam boilers when they are in process of construction is quite as important as periodic inspection when they are in operation."

After keeping the boiler out of commission about three months, repairs were made, as shown in Fig. 2. A steel manhole form was fitted to the outside of the boiler and drilled to match the old rivet holes, also new holes were drilled through the new form, boiler shell and reinforcing ring, the latter being left in the boiler because it was too large to pass through the manhole. The calking had to be around the outer edge of the new form instead of the inside edge of the manhole, as formerly. This method of repairing permitted all of the rivets to be driven from the outside and it made a good job.

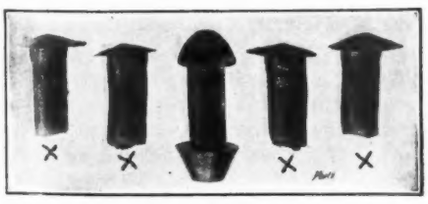


FIG. 3. RIVETS WITH IMPROPERLY FORMED HEADS

It was fortunate that the casting broke at such an opportune time, for if the joint had begun to leak when the boiler was under full pressure and someone had attempted to tighten the bolts, the result would have been disastrous.

J. W. PARKER.

Clinton, Mass.

Trouble with a Barometric Condenser

We are experiencing considerable trouble with a Bulkley condenser installed as shown in the accompanying sketch. The distance from the overflow of the hotwell to the exhaust nozzle is 34 feet and about 1 foot more to the exhaust pipe. The canal gives a head of about 4 feet to the pump.

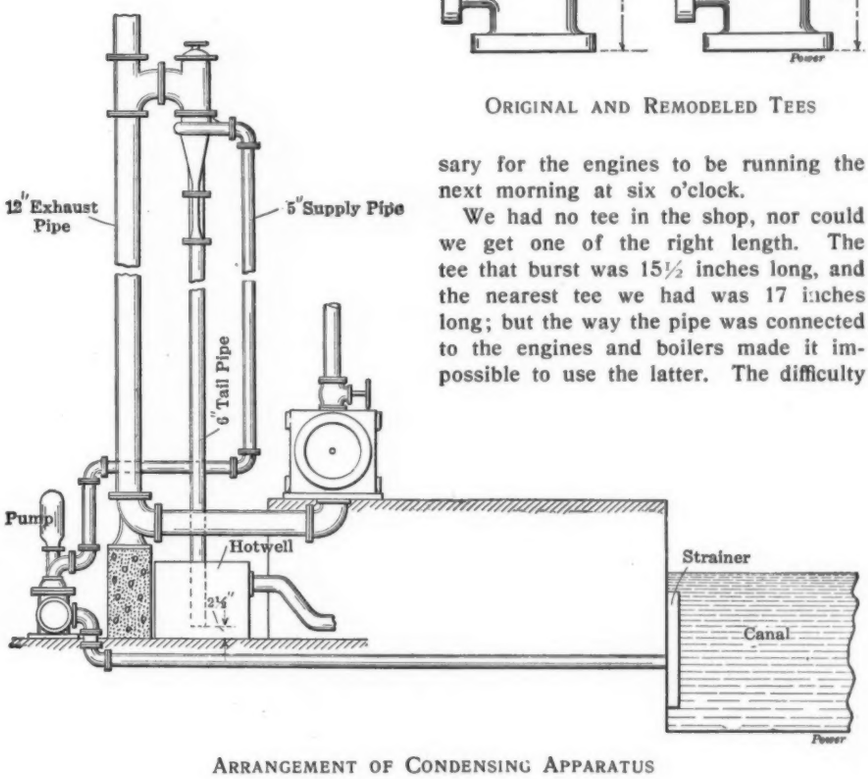
Frequently when building up the vacuum, preparatory to starting the engine, such large quantities of water come over from the condenser into the exhaust pipe that it is impossible to start the engine without danger of wrecking it. Sometimes the water appears just as the engine starts to turn over, in which case we open the drip at the bottom of the exhaust pipe to break the vacuum, and large quantities of water are forced through the relief valve. We have no trouble after the engine gets up to speed, nor when shutting down. After the engine has stopped it seems as though the pump could be kept running indefinitely without any water coming over. When the water does come over upon starting up, the level of the water in the hotwell drops very low.

We tested the condenser at 50 pounds for leaks and found only two small ones. These were repaired but it made no difference. Our vacuum holds quite steady at from 24 to 26 inches. An expert from the condenser manufacturers was sent to investigate the trouble but at the time he was here the condenser worked perfectly.

The engine is a simple 26x60-inch Corliss and has been in operation for about 20 years without giving any previous trouble. The pump seems to be in good condition and there is no exhaust-steam heater on the exhaust line. Perhaps some reader of POWER can solve the mystery.

W. P. ELSTON.

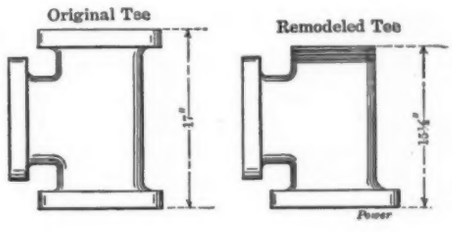
Taunton, Mass.



ARRANGEMENT OF CONDENSING APPARATUS

Remodeling a Tee

One afternoon an 8-inch tee on the main steam line feeding our two engines, burst and it was absolutely necessary for the engines to be running the next morning at six o'clock.



ORIGINAL AND REMODELED TEES

We had no tee in the shop, nor could we get one of the right length. The tee that burst was 15½ inches long, and the nearest tee we had was 17 inches long; but the way the pipe was connected to the engines and boilers made it impossible to use the latter. The difficulty was overcome in the following manner:

The 17-inch tee was placed in a lathe and 1½ inches cut from one end, the remaining portion of that end being turned round and smooth. A standard 8-inch pipe thread was then cut on it (see sketch) and an extra-heavy 8-inch pipe flange was fitted. This gave us a 15½-inch tee which served the purpose as well as a solid tee, and inside of six hours we had the engines running again.

E. W. NEAL.

Pittsfield, Mass.

Preventing Boiler Explosions

In a recent editorial entitled "An Epidemic of Boiler Explosions" you say, "steam users will either have to submit to Governmental inspection of their boilers, have them insured with competent inspection, or employ engineers who are qualified to judge of the safety of the apparatus in their charge, make them responsible for it and comply with their advice and requirements." If either proposition were accepted there would be fewer boiler explosions, and if there were a cold-water sprinkler which would automatically admit water into the steam space when the pressure was too high, all would be well.

PETER VAN BROCK.

Jefferson, Ia.

Supplying Air to Elevator System

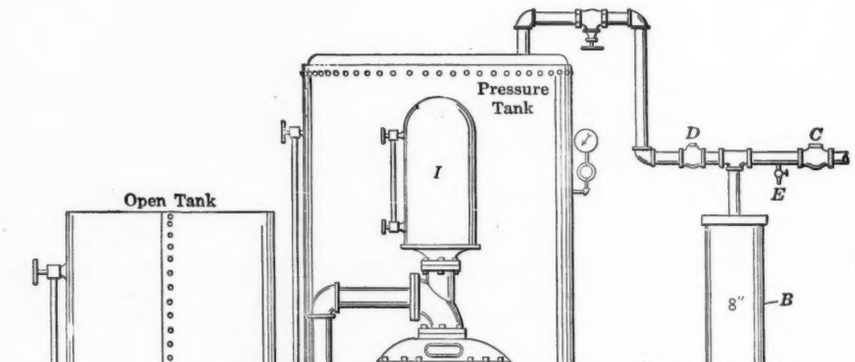
The accompanying sketch shows an ingenious method of forcing air into the pressure tank of an hydraulic-elevator system. The cost of installing this device is small as compared with that of an air compressor, and it gives very satisfactory results.

The method of operation is as follows: First, valve *F* is opened while the pump is in operation and when *B* becomes full petcock *E* will show water. Just before air begins to be sucked in, on the return stroke of the pump, *E* is closed; this draws air in through check valve *C*. On the next stroke of the pump, the pressure closes *C* and forces valve *D* open, and the air passes over into the pressure tank. By this arrangement, no air gets into the body of the pump, the reservoir *B* being filled with water and pipe *F* is not sufficient to drain it at one stroke of the pump.

The pipe and check valve at *G* is for charging the air chamber *I* which loses air through the gage valves. By closing valve *H*, air is drawn in through *G* and soon appears in the top of the glass on the air chamber.

A. C. WALDRON.

Revere, Mass.



PIPING BY WHICH PUMP IS MADE TO SUPPLY AIR TO SYSTEM

Leaky Floats

I would be glad to have some POWER readers offer suggestions as to how to repair leaky floats.

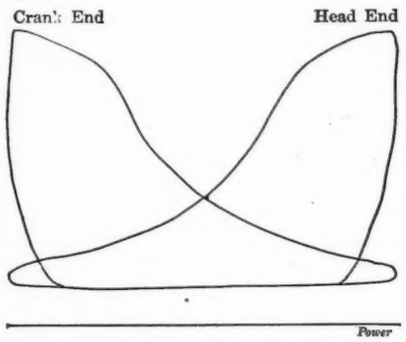
I have seen solder used but it would not be long before the floats, so repaired, were leaking as badly as ever.

E. GRANFIELD.

Boston, Mass.

Diagrams for Criticism

The accompanying diagrams were taken from a 28x36-inch Murray engine running at 120 revolutions per minute. The steam pressure at the throttle was 150 pounds and the scale of the indicator spring was 60 pounds to the inch. Will some readers kindly tell me where the cards can be improved?



DIAGRAMS FROM A MURRAY ENGINE

J. D. ROBERTSON.

Battle Creek, Mich.

Exhaust Steam for Heating Water

It occurs to me that engineers, as a rule, do not realize the saving that can be effected by using the heat contained in the exhaust steam and the discharge from steam traps. It is a common practice to use exhaust steam for heating boiler feed water, but for this purpose only a small amount of the heat contained in the exhaust is used, the remainder being allowed to escape into the atmosphere. In many plants where exhaust steam is used for heating during cold weather, it is wasted into the atmosphere during the summer months.

It is a simple matter to condense the exhaust and return it to the boilers. There are two benefits derived from this; first, the amount of scale formed in the boilers is decreased; second, the water bill is less.

Where exhaust steam is used in heating water for house service, 2½ square feet of cooling surface per horsepower is ample for condensing the steam. A tank 42 inches in diameter by 10 feet long, containing a brass coil having a total surface of 140 square feet with a 2½-inch cold-water inlet and a 2-inch hot-water outlet, should heat water enough for the use of about 700 people; the exhaust from a 50-horsepower engine together with the discharge from steam traps being used to heat the water.

I utilize the exhaust steam from our engines and the discharge from the steam traps to heat all the water for bathing, laundry and cooking purposes in a large institution with which I am connected.

H. B. BRAND.

Brooklyn, N. Y.

A Hydrostatic Test

Sometime ago at a marine-engine works in England, a hydrostatic test was applied to a number of small pinnace boilers. The intended working pressure was to be 150 pounds per square inch and the stipulated test pressure was 300 pounds. The hydraulic service at the works is under 1500-pounds pressure and the method of testing the boilers was to connect them to the hydraulic main, and by partially opening the valve, allow the required pressure to accumulate in the boiler. This plan usually worked all right but on this particular occasion the valve stuck and the pressure in the boiler rose to 1500 pounds. The valve was finally closed and after relieving the pressure, we made a careful examination of the boiler without finding any leak or serious deformation. The remarkable part of it was that when the boiler was installed and steam was gotten up, it leaked all over and had to be consigned to the scrap heap.

JOHN L. SHELDON.

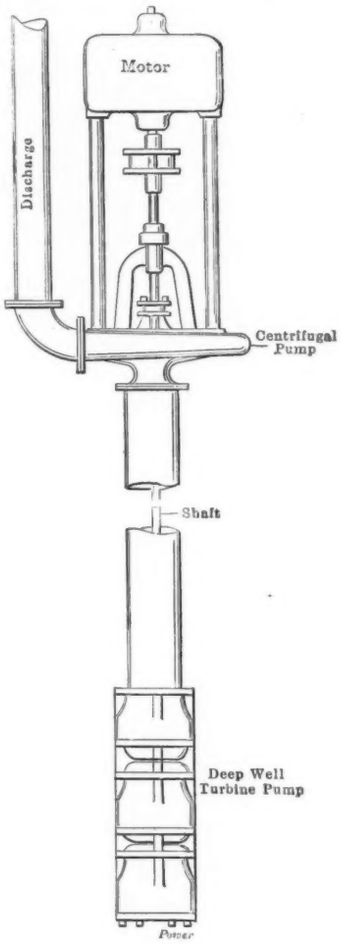
Prince Albert, Can.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues.

Will the Pump Raise the Water?

In the July 5 issue W. F. Callister asks if, when the water level in a well is 70 feet below the surface and an impeller is used to raise the water to within 10 feet of the surface, a pump will lift the water from this point. This has been done. In 1904 there was a pump



COMBINED AUGER AND HORIZONTAL CENTRIFUGAL PUMP

installed in an orange grove in California, which consisted of a combined auger and horizontal centrifugal pump. This pump was designed to draw 1000 gallons of water per minute from 75 feet below the surface and raise it to 80 feet above the surface. This arrangement is said to be patented by Charles D. Boydston.

At the present time there is in the course of erection a pump of similar design; the deep-well part of the pump is of the turbine type and the surface part is a motor-driven vertical centrifugal pump. The sketch shows the arrangement of this apparatus. The impellers of the turbine pump and the vertical centrifugal pump are mounted upon a common shaft. Both pumps are driven by means of a vertical motor through a flexible coupling.

In Mr. Callister's case there is no question as to the ability of this scheme to do the work as the theoretical suction distance for a pump is 34 feet at ordinary altitudes, although, due to imperfect vacuum and water vapor, it actually is somewhat less than 30 feet.

F. E. SPERRY.

Aurora, Ill.

With reference to Mr. Callister's pump problem, as described in the July 5 issue, there is no doubt but that the pump would raise the water the remaining distance. The water is raised to a level of 60 feet above the supply by an impeller and as far as the pump is concerned this is its supply level and the pump should be able to create enough suction to raise the water some 25 feet higher provided the impeller does not prevent the transmission of atmospheric pressure through the water.

JOHN FRENCH.

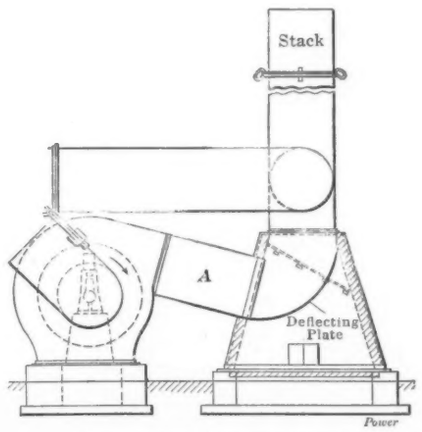
Washington, D. C.

Induced Draft Trouble

I wish to suggest that the trouble with Mr. Faulthier's induced-draft installation, described on page 1263 in the July 12 issue, is caused by the smoke being deflected downward on striking the sloping wall of the chimney base and by the eddies that are set up. If a curved deflecting plate were put in from the lower edge of pipe *A* and curved to as long a radius as possible, most and perhaps all of the trouble would disappear.

H. B. BRAND.

Brooklyn, N. Y.



SUGGESTED IMPROVEMENT

With reference to the article having the above title, in POWER of July 12, the trouble with the original outfit was twofold.

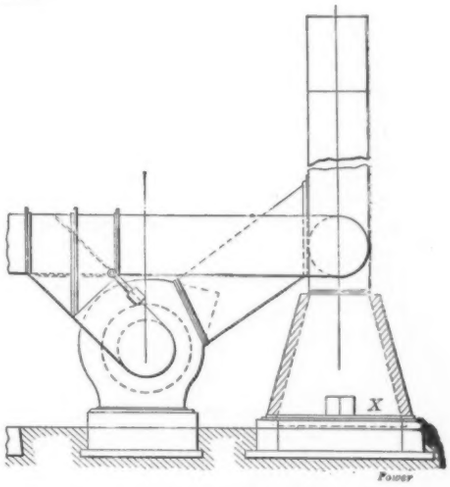


FIG. 1. NEW ARRANGEMENT OF FAN OUTLET

First, the engine was too small for the duty.

Second, the discharge connection from the fan to the stack could not possibly be of a worse design. The gases are discharged into one corner of the stack

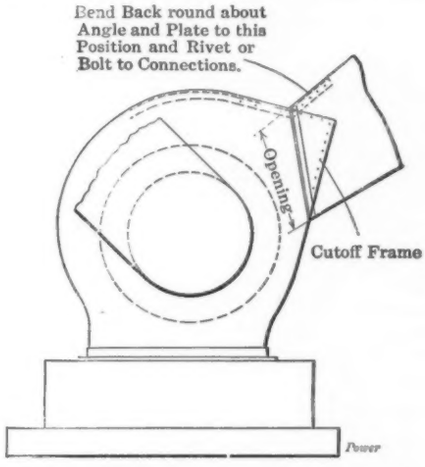


FIG. 2. DETAIL OF FAN ALTERATIONS

base, which forms a pocket. The impact reverses the gases and causes them to eddy.

The cheapest remedy is to install the 9x14-inch engine and alter the discharge connection so that it discharges the gases into the stack at a level not lower than the natural-draft connection, as shown in Fig. 1. This can be accomplished with little expense as it only means removing the outlet frame and a short portion of the roundabout plate. Details of this alteration are shown in Fig. 2.

E. W. SMITH.

Worcester, Mass.

Breakage of Follower Bolts

Mr. Handley's follower bolts, as related in the June 28 number, broke so soon after setting them up that it looks as though it may have been a case of main strength applied at the end of a 24-inch wrench. With a wrench of that size, and we often see one used, it does not require a very strong man to twist off a 1-inch follower bolt. In this case, although the bolts were not actually twisted off, the beginning of a break may have been made and the jar of a few hours' operation finished the job. A 14-inch wrench is the largest that should ever be used for this kind of work; and it is not a good idea to try to bend the wrench either. For smaller sizes, a 10- or 12-inch wrench is large enough.

After putting in the new piston, Mr. Handley should have disconnected one end of the connecting rod and moved the piston till it struck at each end of the cylinder, marking these striking points on the guide. Then, after connecting up again and finding each center mark on the guides, he would have known what clearance he had, and would not be wondering if the piston were striking. Steam should not have been turned on until this had been done.

WILLIAM E. DIXON.

Hudson, Mass.

Withholding the Facts

In the July 12 issue appears an editorial headed "You Will Never Know," commenting upon a boiler explosion at the Midvale Steel Works.

It states that management of the steel works refused to give any information to POWER'S representative as to the cause of the explosion and the resultant death of two employees.

To me this seems to be a stand which should be impossible to maintain by anyone. In some States the officers of the Midvale Steel Company would very soon be taken from their high perch, especially in Massachusetts.

In a case like this we have two departments which can and do make thorough investigations and their reports are public documents open to all. I refer to the Boiler Inspection Department and the Board of Medical Examiners.

The Boiler Inspection Department has the right to "enter any premises upon which a boiler or engine is situated, and any person who hinders or prevents or attempts to prevent any State boiler inspector from so entering shall be liable to the penalty as specified in this section." The penalty referred to is fine or imprisonment or both.

When the State boiler inspector arrives, and he is usually "Johnny on the spot," he gets all of the details fresh from the men who were present, and the report he submits to the chief of the department is such that anyone can understand, and is as near to the truth as it is possible to get.

Should an accident to a boiler involve the loss of human life, the Medical Examiners step in. These examiners are men of high standing in the medical profession; they use their power, which is great, in case of homicides, suicides or accidental deaths, for the benefit of the commonwealth.

In some large mills life is held to be a cheap commodity. It is proper to bring these matters before the public whenever occasion permits, so that public sentiment may be aroused to secure the establishment of proper authority for investigation.

ALBERT S. SMITH.

Boston, Mass.

Inspection of the Governor

While reading an account of a recent flywheel disaster, my thoughts became centered upon the question of flywheel insurance and the conditions governing it. Looking at the question fairly and squarely, the flywheel protection is somewhat of a misnomer. While admitting that the question of rim velocity and the factor of safety for the various stresses should be thoroughly looked into, to my mind it is really of secondary importance. The governor is more important and should be looked after and insured, as it is the controlling feature of the whole machine. We read of many cases wherein the failure of a governor to act promptly has been the cause of disaster.

While engaged in visiting several engine rooms I ran across one when the engineer was in the act of mending his governor belt while the engine was running; the temporary regulation being made by throttling the steam to suit the load. During the operation of fixing the belt the engineer was temporarily called away to attend to some minor detail, and the thought came to me of what would be the result should the greater part of the load be suddenly thrown off by the breaking of a belt, or the throwing out of a circuit-breaker during his absence. Of course, this does not revert to the insurance of the governor, but the fact remains that insurance of the flywheel would not help matters any. Take the case of a Corliss engine the regulation of which is affected by the action of the governor on the cutoff. Suppose that from improper adjustment or wear the governor refused to act when the balls were rotating in a higher plane than usual and the load were suddenly reduced; the result can better be imagined than described.

A case is related in the July 19 issue of POWER, in the article by G. Trube on "Runaway Engines," in which he states that "Owing to low steam on a Corliss engine the throttle had to be opened." This should have been done anyway as it is impossible to obtain a good full initial pressure and good economy by a partly closed throttle valve. He also says that "Shortly after the steam was raised the speed of the engine became dangerously high and the engineer ran to close the throttle, but could not do so." The question I would ask is this, "What was the matter with the governor that it did not take care of the increase in speed?" If engines are to be governed by throttling it is useless for engine builders to spend valuable time trying to improve and bring the sensitiveness of the governor down to the finest possible point, as it is by this means that economy in the expansion of the steam is produced. Therefore I reiterate that it is governor insurance that is needed more than flywheel insurance.

CHARLES H. TAYLOR.

Bridgeport, Conn.

Increasing the Power of an Engine

The question of "Increasing the Power of an Engine" which was discussed in the July 19 issue of POWER, is a practical one, asked by many examiners in Massachusetts. It seems that the examiner wishes to draw out the ideas of the applicant on how he would know when his engine had reached the limit of its power. My answer to this question is as follows:

The limit to the range of cutoff by the governor being two-fifths of the stroke, the valves must trip at this point if they trip at all. To make the engine do more work, shorten the rod connections to the steam valves. This will give more port opening at the beginning of the stroke. To overcome the objection of too much lead, the eccentric should be moved back. The position of the eccentric, if it leads the crank, would be about 90 degrees in advance; after the change in the rods, connecting the wristplate and the steam valves, had been made.

Very little margin for such a change can be made on a single-eccentric Corliss engine without sacrificing compression or late release. But the double-eccentric Corliss engine can be made to cut off as late as half stroke, and even more work can be gained by shortening the

steam-rod connections to the wristplate and by moving the eccentric back from 90 degrees in advance of the crank. This may be carried to the point where the throw of the eccentric, having reached its limit, would have to close the valve on its return stroke, allowing the steam to be partially forced out of the cylinder as the piston returned. This would give a Corliss valve the same economy in operation as a slide valve, and the engine would not cut off at all; being overloaded the valve would not trip. The engine would be doing much more work than it did with the eccentric set at 90 degrees plus the lap and lead with the cutoff by the governor ranging from one-fifth to two-fifths stroke. This is the change the engineer is expected to make before adding a condenser or compounding or changing the size of the flywheel.

R. A. CULTRA.

Boston, Mass.

Chimney Design

In the issue of July 5 appears an article on "Chimneys and Draft" by John S. Leese, in which the author, under the subhead, "Types of Chimneys," states that "the wind pressure in designing chimneys may be calculated at 56 pounds per square inch."

Whether he wishes us to believe that this is a feat within the power of men, or simply may be the practice among designers, is hard to guess, although I am inclined to the latter view on account of his obvious unfamiliarity with this subject, as shown by this as well as other statements.

In most quarters, and I say this after considerable experience in designing chimneys, a chimney designed to withstand a wind pressure of 50 pounds to the square foot is as securely rooted to the face of the earth as the most extraordinary occasion would require.

H. R. AUSTIN.

Oshkosh, Wis.

Centrifugal Pump Characteristics

In the issue of July 5, on page 1211, in an article on "Centrifugal Pump Characteristics," there appears, in paragraph four, this statement: "The efficiency which at zero head is zero." This is not borne out by the curves and data shown.

At zero head the table shows that 600 gallons per minute were moved, with an expenditure of 400 brake horsepower.

This means that 600 gallons per minute were not lifted to any hight, but were moved at a certain velocity, with the natural result that 400 brake horsepower was expended, or two-thirds of the maximum. If Mr. Sanford will give us the power demanded to produce the 400 brake horsepower, at zero head, the efficiency of the pump at zero head can be obtained. Otherwise, the curves and data are misleading as regards zero head.

Evidently, the pumps cannot deliver 400 brake horsepower, and have no efficiency, which is what the reading matter says, and the curves do not show.

H. G. MANNING.

Pittsburg, Penn.

Long Runs of Pumping Machinery

In the July 19 issue, Mr. Dreyfus asks for information regarding "Long Runs of Pumping Machinery." I have charge of three cross-compound Corliss pumping engines, built by the Laidlaw-Dunn-Gordon Company. Each is 16 and 30x11x30-inch and raises water 247 feet for irrigation purposes. No. 2 engine has run continuously for over 56 days and No. 1 for 42 days, each carrying full load two-thirds of the time.

R. A. DAVIDSON.

Colton, Cal.

Increasing the Capacity of Boilers

I was very much interested in T. T. Parker's letter in the July 19 issue on "Increasing the Capacity of Boilers," especially that portion concerning vertical fire-tube boilers. This type of boiler has many features of merit over other types of either fire- or water-tube boilers, with only one objection, that is, the removal of scale. But this, in itself, is not by any means insurmountable, even with bad water.

One feature of this type of boiler that does not exist in any other type is the fact that every tube evaporates the same amount of water, owing to the hot gases being evenly distributed over the tubes. In the horizontal tubular boiler, the three upper rows evaporate far more water per square foot than the lower rows and only the upper half of each tube is available as a positive heating surface. In the water-tube boiler there is a very high rate of evaporation at one end of the lower tubes, and as a result these tubes give trouble by bulging and bagging. This may be charged to a poor distribution of the heat, the high temperatures being applied locally to a very small portion of the total heating surface. It is true that in the vertical fire-tube boiler, the greatest heat is first applied quite a distance from the steam space, but in practice this amounts to little as the temperatures would be equal in the water contained above the grate line and we have superheat to dry the steam.

The theory of having the entering feed water meet the products of combustion just prior to escaping to the chimney is of little practical value, for, in a steam boiler, with a well-defined circulation, the temperatures are practically equal where the water surface is exposed to the products of combustion.

Mr. Parker refers to vertical boilers without water legs, a type I have never met with. I would consider a water leg valuable in permitting the lower tube sheet to be flanged downward, thus facilitating the cleaning of the sheet and the collection of loose scale at the mud ring below the grates. In addition, by using hollow staybolts with the holes 3/16 or ¼ inch in diameter, we would have the admission of air above the fire that is necessary for proper combustion. This would be properly distributed and diffused equally to all parts of the furnace, which is not the case with other types of boilers. The air would enter through very small apertures and absorb heat from each hot staybolt. Consider a vertical boiler with a 72-inch lower tube sheet having a 4-inch furnace, with staybolts on 6-inch centers; 266 bolts would be required and with 3/16-inch holes one would have about 7 square inches of air inlet, granting the inlets were free. One could easily determine the proper admission and then block off the inlets not needed, doing this from the furnace sides of the staybolts. But the value of such air admission in completing combustion before the gases enter the tubes appeals to me and is a special feature of this type of boiler provided with a water leg.

The boiler would probably require more care, as regards the admission of cold air when cleaning the fire. The tube ends would be susceptible to such mistreatment as in a locomotive boiler and tube leakage would be in proportion to the care the boiler received. Men accustomed to leaving a fire door open without taking the trouble to close the damper would, no doubt, spend far more on repairs than others. The personal equation would figure largely, but with proper care, tubes should give no more trouble in this type than in other types.

As the shell is out of the fire and is subjected only to temperatures equal to that of the water and steam, its life should be much longer than that of a horizontal tubular boiler. Properly jacketed outside, the inside could be painted with graphite, or other good paint, thus eliminating internal corrosion. Without internal corrosion and with no wear and tear on the shell, the annual depreciation would be greatly decreased.

With a horizontal tubular boiler of 150 horsepower I would assume the annual cost of maintenance to brickwork as $50, or $1000 for 20 years—the average life of such a boiler; this is equal to 50 per cent. of the first cost. But this brickwork is of no value to the steam user except for jacketing the furnace and boiler. It

is in a sense a flywheel in only the best furnaces and is inefficient in proportion as it permits air leakage beyond the bridgewall or over the tubes, where such leaks are extremely wasteful. Bricks are porous to water and much more to air or gases. With the closure of the fire door in a vertical fire-tube boiler, provided with a water leg, there are no such deleterious air leaks, no first cost for brickwork and no repairs of this nature, that in 20 years aggregate one-half the cost of the initial installation. The amount of water evaporated per pound of combustible is practically the same in all types of boilers.

A. J. FORAN.

Paterson, N. J.

Boiler Failures

The safety of a boiler demands closer attention at the present time than ever before, from the fact that the majority of our cylindrical boilers are now made of steel plate. In practice this plate shows that it does not deteriorate evenly, and that crystallization is more liable to occur in the steel plate than in the old charcoal-iron plate, from which boilers were formerly made. As a result, a boiler that is apparently sound may have a hardened spot which will snap like glass if subjected to an uneven stress. This condition is often found on the flange of the flue head, or on the rear head of a return-tubular boiler.

We find many boilers in which the ends of the tubes have partly rusted away. Now, a tube may be considered as a stay, and it will hold in exact proportion to the shear on the bead; therefore, when the beads are gone, the stress comes on the diagonal brace supporting the head. Frequently this brace is in a corroded condition, or perhaps the base of the brace is cracked and the slot has become worn until the brace pin becomes loose, allowing it excessive play. A loose brace pin is a dangerous thing for any boiler, and especially if the boiler is otherwise weakened. To illustrate this point, consider an elevator cable which will hold a car in ordinary service, but should the car drop a few stories, the cable would probably snap. This is true of a boiler head if the flues are rusted away and the brace has play; a slip of the head is apt to cause an explosion. With the brace pin tight, this would not happen.

This condition is more pronounced at the top head on upright boilers and especially so on boilers that have no hood over the smokestack to prevent rain from coming down. A wire brush should be used on the tube heads to remove the scale, in order to see the exact condition of the tube sheets. Many of the accidents are from small boilers that have no regular inspection. Merely looking over a boiler does not mean that it is receiving a careful inspection. Brace cracks are found only by close observation and often a brace foot will be almost gone before the crack can be easily seen.

C. R. MCGAHEY.

Sheffield Ala.

Bypass on Boiler Stop Valves

The letter of James W. Little, in the July 26 issue, came as a surprise to me, as I was under the impression that all well designed plants used the bypass on their stop valves.

In 1898 I designed and installed a small high-pressure plant in the South, and in this instance each of the two 300-horsepower Babcock & Wilcox boilers was connected to the main header by an 8-inch Walworth rising-spindle, heavy-pressure, flanged gate valve, on which a bypass valve of similar construction, of about 1¼-inch diameter, was placed.

These valves and all other pipe flanges on the system were packed with ring gaskets of asbestos millboard, soaked in water and then drawn down until the flanges were practically together. These valves and connections gave no trouble during my stay at the plant and I have heard of none since.

I thought at the time that I was keeping in line with modern practice by installing those bypass valves, instead of violating established precedent, but now Mr. Little mentions it, I am reminded of the fact that no other plant where I have worked has been thus equipped. Whether this state of affairs is caused entirely by precedent, or largely because of first cost, would be an interesting subject for discussion.

I still believe it to be a good investment if only as a matter of insurance against carelessness or inexperience.

L. L. ARNOLD.

East Greenwich, R. I.

Vibrations in Boiler Header

Referring to a letter by G. E. Miles, under the caption "Vibration in Boiler Header," in the issue of July 19, it appears to me that the vibrations were started from the boiler which had not been cut out from the line. If he had closed the main steam valve on the boiler from which the fire had been drawn, the vibrations would have ceased immediately.

Mr. Miles did not state how long the fire had been withdrawn from under the boiler when the vibrations began. There is, no doubt, some peculiar action which goes on in a boiler from which a fire has been withdrawn previous to closing the main steam valve.

I believe that the opening and closing of the steam valves in a cylinder cause the steam in the header to surge forward and back with great rapidity. The boilers which have a fire under them generate enough steam to run the plant and at the same time, on account of the continuous generation of steam, they are in a condition to resist the surging of steam, just mentioned. This is not so with the boiler to be cut out; the ebullition of the water has ceased and no more steam is being generated; consequently the boiler is subjected to a succession of rapid shocks which would cause the header to vibrate if other conditions were favorable.

Mr. Miles states that the vibrations slowly but gradually increased. This is just what would be expected, and the engine running under a light load tended to increase the intensity of the vibrations on account of shorter cutoff.

I recall an incident that happened several years ago, which would tend to show how steam acts under such circumstances. It was in a steel mill which had waste-heat boilers located over the puddle furnaces. The puddlers had drawn the heat and shut off the steam blast. The steam pressure in the other boilers, all of which were in operation, was raised to the blowing-off point; and the safety valve began to hammer on the boiler from which the puddlers' heat had been withdrawn. The men became frightened and began to run in every direction, thinking the boiler was going to explode. The assistant master mechanic climbed up on top of the boiler and attempted to screw down the adjusting nut on the stem, in order to stop the hammering. It hammered so much that he could not get the wrench on the nut and had to give up. After repeated efforts by others to stop the noise, all of which were failures, it finally occurred to someone to close the main steam valve. This was done and the trouble ceased instantly.

W. W. SCOTT.

Youngstown, O.

Packing Condenser Tubes

The method of packing condenser tubes as described in Mr. Gibson's article in the July 19 issue, I have found to be unnecessarily slow and tedious.

A much better method is to use an old piece of tube, one end of which is cut to fit between the condenser tube and the stuffing box, and the other end to fit a carpenter's brace. With packing cut into lengths of about 18 inches and having been previously soaked in boiled oil, a perfectly tight job can be easily obtained in a very short time.

W. E. SIMPSON.

New York City.

The Mechanical Engineers Abroad

The "Celtic," bearing a party of 148 members of the American Society of Mechanical Engineers and their families, dropped anchor in the Mersey, early on Sunday evening, July 24, and was shortly boarded by the deputy lord mayor of Liverpool (John Lea), President J. A. F. Aspinwall of the Institution of Mechanical Engineers, Professor Watkinson, of the University of Liverpool, and others. The three gentlemen named extended the welcome and greeting of the city, the Institution and the university to the visitors. Professor Watkinson spoke very kindly of his treatment in "The States"

FIG. 1. JAMES WATT'S HOUSE, HEATHFIELD HALL, AT BIRMINGHAM

last year and complimented our universities, that of Illinois especially, upon their research work. The addresses were responded to by Dr. W. F. M. Goss, dean of the University of Illinois and vice-president of the American society.

Most of the party remained aboard Sunday night and left Liverpool on a special train composed entirely of dining cars, at 11:15, arriving at Birmingham somewhat after 2 p.m. The afternoon and evening were spent in getting settled, meeting old and making new acquaintances. Handsome international badges enameled in red, white and blue were distributed.

The first session of the meeting was called to order in the lecture room of the Birmingham and Midland Institute by the right honorable the lord mayor of Birmingham, Alderman W. H. Bowater, who said that it was quite fitting that the Institution should bring its guests to Birmingham, for it was here that the Institution was founded at the Queens hotel, over sixty years ago. He extended a hearty welcome in the name of the city of Birmingham and called upon George Tangye, who presented to the American society an autograph letter written by James Watt to his partner Matthew Boulton in 1777, a photograph of which is reproduced herewith. The addresses of welcome were responded to and the gift accepted on behalf of the American Society of Mechanical Engineers by Doctor Goss.

President Aspinwall, upon taking charge of the meeting, suggested that the regrets of the meeting for his inablity to be present be telegraphed to President Westinghouse, of the American society. This was done and after approving the minutes of the last meeting of the Institution and electing 122 new members, the meeting passed to the consideration of the professional papers. These dealt for the most part with railway practice, and such portions of the meeting as will interest POWER readers are fragmentary, as F. H. Clark's statement that out of over 1000 locomotives they have about two failures per day sufficiently serious to incapacitate the engine, and that of these failures 40 per cent. are due to leaking tubes and 5 per cent. to burst tubes. Lunch was served at the town hall.

In the afternoon, excursions were made to Dudley Port, Tipton, where are the South Staffordshire Mond Gas Company's works and the testing station of the Pump and Power Company, of which H. S. Humphrey is the managing director, and where the famous Humphrey internal-combustion pumps were on view; to several industrial establishments and to Stratford-upon-Avon, the waterworks, Worcester, Stoneleigh Park and Kenilworth.

Your editor spent a delightful after-

FIG. 2. WATT'S WORKSHOP AT HEATHFIELD HALL

noon at Heathfield, the residence of James Watt in his later days and the present home of George Tangye. The original part of the house is at the right of the accompanying picture, and the double windows showing through the trees in the upper story are those of "the Watt room," the photograph of which is also reproduced, where the master had his little foot lathe and worked out his later inventions. Watt was not only an engineer but a diplomat, as a review of his correspondence shows. Mr. Tangye says that it is also established that he made sun pictures long before Daguerre. The interior of this room, with his belongings and materials as he left them—oil and chemical in bottles and jars, a handkerchief where his head rested on the bar of a reproducing machine, wax tapers and lots of little personal effects—is kept by Mr. Tangye as Watt left it, full of tradition, of memories of the Lunar Society, whose membership, including Josiah Wedgwood, Matthew Boulton, Dr. Erasmus Darwin and Doctor Priestley, met here on moonlight nights,because they could not see to come otherwise, and of leading strings back into the beginnings of the age of power. And the genial and kindly present owner, glad in the

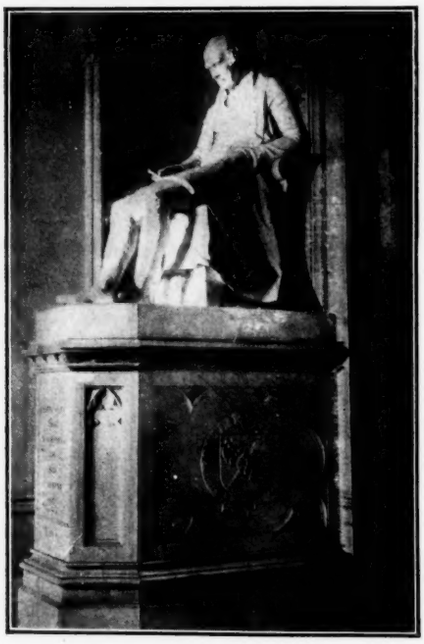


FIG. 3. MONUMENT OF WATT OVER VAULT WHERE HE IS BURIED

possession of it, pleased to be in a way a part of it, for his family is related with the Trevithicks and his own house and name are prominent in the more recent history of engineering, fits charmingly with the picture and finds his greatest pleasure in sharing it with those who appreciate its memories and significance.

Across the fields is St. Mary's church, of Old Handsworth, where James Watt is interred. The accompanying photo-

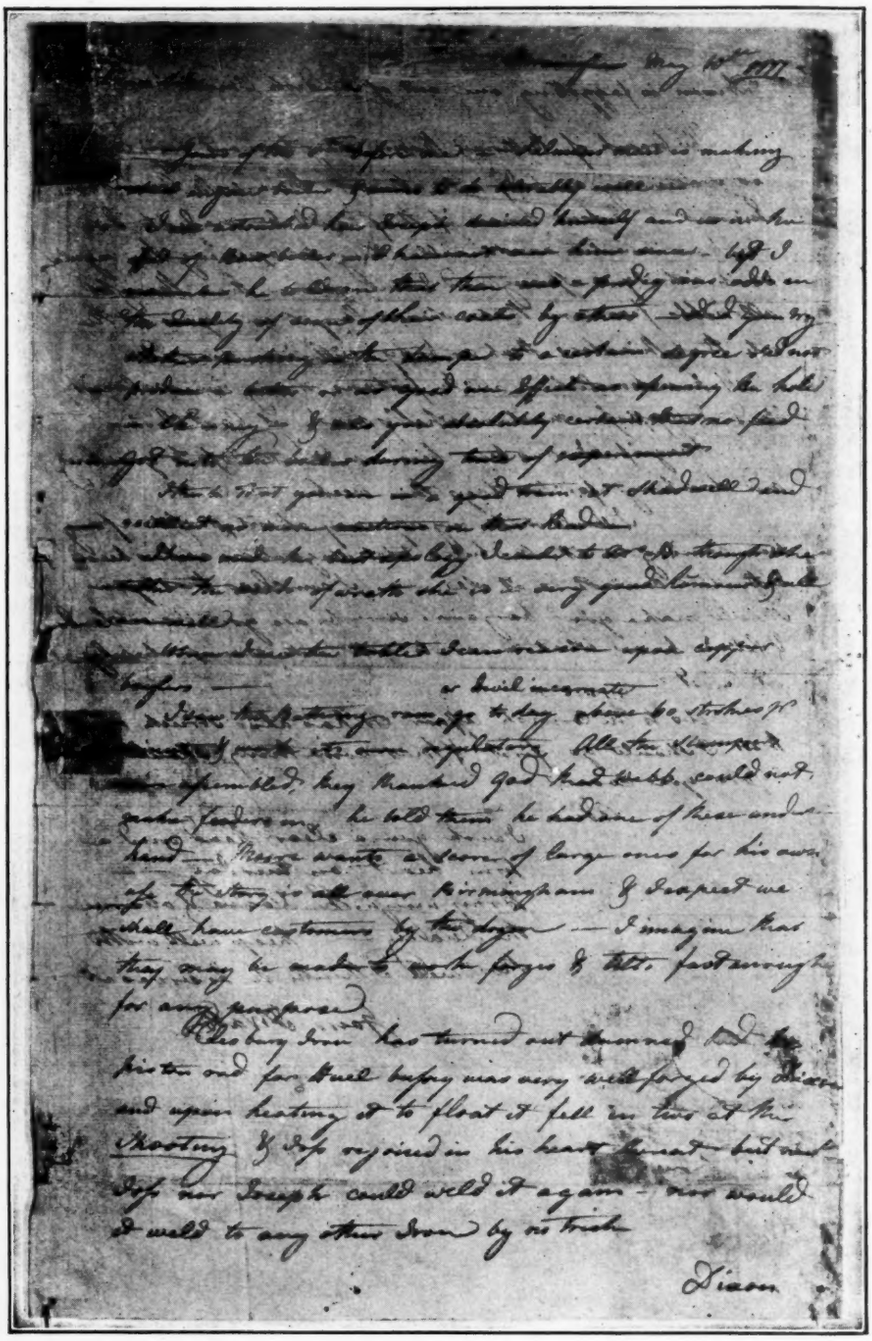


FIG. 4A. WATT'S LETTER TO MATTHEW BOULTON

graph shows the monument above the family vault. At the right of the altar is a bust of Matthew Boulton, and at the left one of William Murdock, the inventor of illuminating gas. In the Gillott pen factory nearby is a Watt engine which has been running for 98 years and is still at work.

For kindly interest and attention during my visit to Heathfield Hall, grateful acknowledgment is made to R. H. Kirton, curator of the Watt museum.

In the evening there was a garden fete at the Edgbaston Botanical Gardens, at which the lord mayor, his wife, and Neville Chamberlain, Esq., chairman of the executive committee, received and the band of the Portsmouth division of the Royal Marines furnished the music.

Wednesday morning was devoted to the consideration of papers, the afternoon to a visit to the University of Birmingham, or to Mitchells & Butler's brewery for those who preferred, and in the evening a reception was tendered in the Council House by the lord mayor and his lady.

Thursday was devoted entirely to excursions, of which choice was invited from a number of the principal attractions which these historic and industrial Midlands afford. On Thursday evening the party left for London, arriving in time to take part in a conversazione at the house of the Institution of Mechanical Engineers.

Grooving in Steam Boilers

By W. H. Booth

When pressure is raised in a boiler with longitudinal lap seams the outer plate tends to bend over the edge of the inner lap as the plates strive to become truly circular. The bending stress causes grooving of the plate. By proper precautions this grooving may be avoided.

Every vessel containing internal pressure tends to assume the form of a perfect sphere. Vessels such as steam boilers are restrained to shape by means of stays to their flat surfaces. The staying of the ends of a cylindrical boiler enables the barrel or body of the boiler to strive only to maintain a truly circular form in cross-section. The cylinder form is up to a point as scientific as the spherical, but it introduces the difficulty of the stayed flat end. The old egg-ended boiler once so common in Great Britain has hemispherical ends and cylindrical body, and is therefore exempt from any tendency to change its form, for both ends and body are ideal in shape.

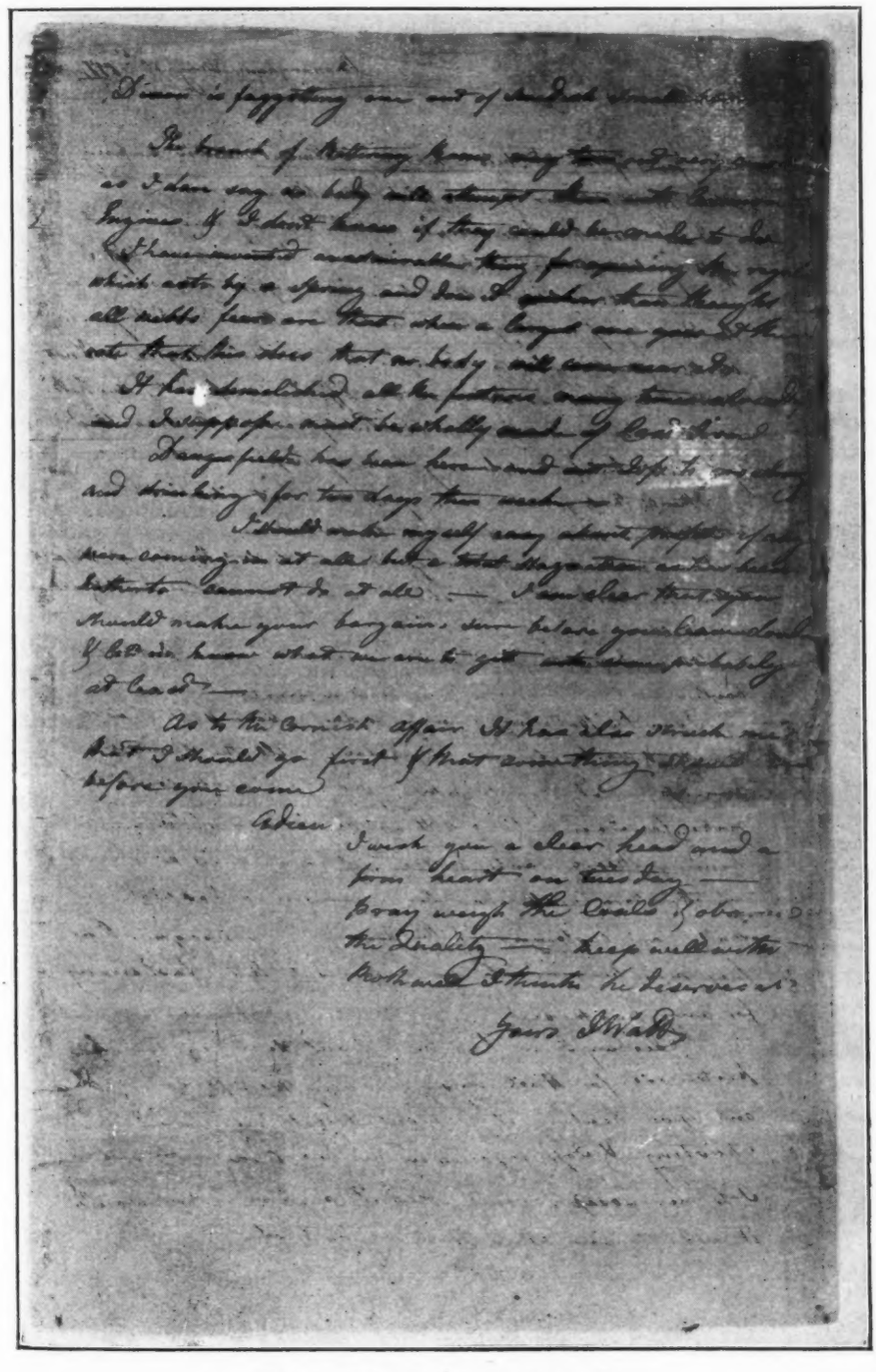


FIG. 4B. WATT'S LETTER CONTINUED

If a plate be bent into a circle and the ends laid together and riveted, there will not be formed a true circle, but when put under pressure this ring of plate will strive to become truly circular, and in so striving there will be a tendency to bend the outer plate over the edge of the inner lap. It is this bending stress which causes grooving of the plate because the sharp nip over the edge of the plate cripples the bent plate along that line and facilitates the action of the water upon its overstrained particles. Steam does not appear to have such an effect, so that it became usual to place the longitudinal seams of shell boilers always above the water line so as to avoid this action of the water.

If a boiler ring be drawn on paper to a true circle the lap joint can be designed so as not to disturb the true circularity. This is done by bending one plate lap at an outward angle and the other equally inward. So shaped, the rivets through the two plates are not exactly radial to the boiler, and there will be no movement of the joint, though the rivets will now be partly in tension and there will be a steady stress along each lap at the angle of bend. There is also a great tendency to injure the plates when bending them to the necessary angle. As a matter of good practice, therefore, the lap weld has been abolished and rings of plate are now made as truly circular as possible, the ends butting together and the joint being made by a pair of cover strips placed inside and outside the boiler. All rivets are now radial and there is no tendency to depart from the initially correct form.

Did boilers work under a steady pressure, there could be no change of curvature once the shell had taken the form of equilibrium. But since the stiffness of the plates is exerted against the steam pressure the boiler will become more or less truly circular as the pressure rises and falls. This variation of pressure is at the root of all grooving troubles of the order referred to, for the plates are always striving to approach their initial form and the pressure is always striving to push them to the exact circle.

Some years ago several lap-jointed boilers on a British railway exploded as a result of grooving along the longitudinal seams. The effect of observation of other boilers and of the exploded boilers brought about, first, the placing of such seams above the water level and, next, of abolishing this type of seam in favor of the butt seam with double butts. The single-butt seam has a stress in it along the middle line of the butt strip. Hence the use of the double-butt strap, each

strip or strap somewhat less in thickness than the plate to be jointed. With this seam double riveted and with a third row of rivets at double pitch beyond, the strength of a joint may be made about 85 per cent. of the strength of the solid plate, and the tension is always through the middle line of the plate thickness, or at the point equally on each side of the middle thickness, so that no tendency to alteration of shape is induced so long as the finished rings are truly circular.

Grooving does not appear to be set up unless the water is slightly corrosive, but very little corrosive influence appears to be required to cause grooving when there is the necessary changing stress in the plate. Flat surfaces are very apt to groove, especially when exposed to a sharp bend. Thus, in the Lancashire type of boiler the front end plate where attached to the furnace tubes is thrust forward ¼ to ⅜ inch by the expansion of the furnace crowns, and the plate often grooves at the point where it is sharply stressed at the edge of the connecting angle iron. This grooving was very severe at the time when the front end plate was stayed by gusset stays and the lowest rivets of the gusset angle irons to the front end plate came as near as 4 inches to the rivets of the furnace angle iron. Today the free space is made 10 inches and the plate is so much less sharply stressed over the edge of the angle iron that grooving has been very largely eliminated. The stress may also be much moderated if, when cold, the front end plate had actually to be drawn inward to meet the furnace angle iron. Thus, if drawn in say ¼ inch and the furnace tube, when expanded by the fire, pushes it out ⅜ inch the net expansion is only the difference of the two amounts, or ⅛ inch, and this represents very slight bending when the free plate is 10 inches.

Grooving will occur in short lengths of only an inch and always where a plate is pulled over the edge of an angle iron or other thickness. This conclusively proves that its initial cause is the overstress of the surface fibers where nipped short over a corner. A study of grooving is a lesson in boiler stresses and is convincing proof of the need for true forms or elasticity. True forms are not always convenient and flat surfaces are very much employed in boiler work with suitable staying. But flat surfaces require very careful design, and the designer will be better fitted for the work if he has observed the defects to which flat surfaces are prone. The whole question of grooving in boilers seems to resolve itself into one of stress in the outer fibers of the plate. If these are too much stressed they appear to be more open to attack by the water. The extreme fiber stress is of course reduced when the bent portion of a plate is long rather than short. Any acidity of the water simply intensifies the action. That these are the causes may be taken as pretty clearly proved by the fact that active grooving has often been stopped by cutting away one or two of the attaching rivets of a gusset stay angle iron, thereby adding to the length of the part of plate under stress and diminishing the angle of bend.

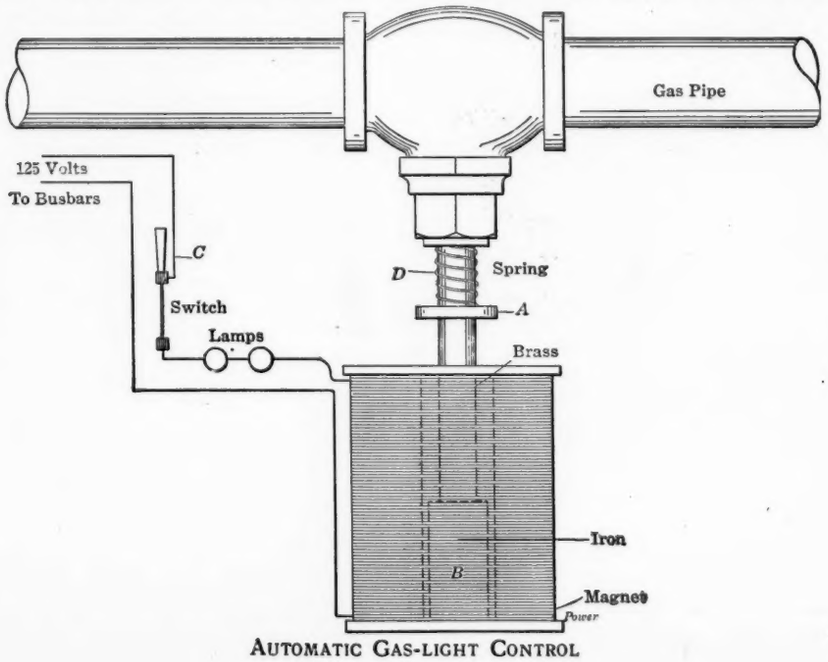
Grooving will take place round a very stiff flange of a manhole mouthpiece on a flat plate and reason would lead one to make all such blocks with flange attachments, as thick as possible, and moreover extended to some considerable extra width by an extension of tapering section finishing quite thin at the edge. This is perhaps an unnecessary refinement though the principle underlying the suggestion might be made useful for some special cases or conditions.

Gas Lighting in Electric Plant

Although no chief engineer cares to admit the probability of affairs ever reaching such a crisis that the entire steam plant will be out of commission, it is nevertheless a wise precaution to provide for some means of illuminating the engine and boiler rooms of his power plant in case the current should fail.

One method is to install a small storage-battery system, to which a number of very small incandescent lights are wired and distributed at suitable points about the engine and boiler rooms, one light always being in circuit and located at the switchboard. This system can be arranged at a small cost, and the battery charged at any convenient time.

Another, and perhaps the better plan where gas is obtainable, is to arrange for a number of gas lights, at a convenient point on the ceiling or elsewhere, fitted with a small gas jet that is always burning, so as to ignite the gas when it is turned on to the larger burners. An automatic method of lighting is shown in the illustration, and is the system installed in a large power station at Birmingham, Ala. The system consists of a magnet, the winding of which is connected to the 125-volt busbars which are always "alive" so long as the machines supplying them are in operation.



AUTOMATIC GAS-LIGHT CONTROL

At a convenient point a globe valve is connected to the gas pipe with the valve stem in a downward position, as shown. The stem has no threads and moves easily up and down when operated. To the valve stem is screwed a piece of brass rod having a collar *A* on one end. This brass rod also screws into the iron plunger *B*, which is suspended in the coil at a point below the center. As the current is always passing through the coil when the switch *C* is thrown in, the tendency is to draw the plunger *B* toward the top of the coil, as the magnet is strong enough to overcome the resistance of the spring *D*, the weight of the valve and stem and the pressure against the valve disk. The valve will open the moment the current passing through the coil fails. This permits gas to pass to the various gas burners distributed throughout the plant, where it is ignited by the small jets in the burners.

Issued Weekly by the
Hill Publishing Company
JOHN A. HILL, Pres. and Treas. ROB'T MCKEAN, Sec'y.
505 Pearl Street, New York.
150 Michigan Avenue, Chicago.
6 Bouverie Street, London, E. C.
Unter den Linden 71—Berlin, N. W. 7.

Correspondence suitable for the columns of POWER solicited and paid for. Name and address of correspondents must be given—not necessarily for publication.

Subscription price $2 per year, in advance, to any post office in the United States or the possessions of the United States and Mexico. $3 to Canada. $4 to any other foreign country.

Pay no money to solicitors or agents unless they can show letters of authorization from this office.

Subscribers in Great Britain, Europe and the British Colonies in the Eastern Hemisphere may send their subscriptions to the London Office. Price 16 Shillings.

Entered as second class matter, April 2, 1908, at the post office at New York, N. Y., under the Act of Congress of March 3, 1879.

Cable address, "POWPUB," N. Y.
Business Telegraph Code.

CIRCULATION STATEMENT

During 1909 *we printed and circulated* 1,875,000 *copies of* POWER.

Our circulation for July, 1910, *was* 138,000.

August 2..... 34,000
August 9..... 34,000
August 16..... 34,000

None sent free regularly, no returns from news companies, no back numbers. Figures are live, net circulation.

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Power and The Engineer

Editorial

Safety of Boilers

It is a well known fact that there are fads and fashions in matters mechanical as well as in spring millinery or the latest cut of clothes. The most recent fad affecting the steam-generating end of power-plant equipment is to try and see how much steam can be generated per square foot of water-heating surface, and some engineers have apparently gone "daffy" on the subject. It seems that since the results of certain experiments made at St. Louis to prove or disprove Prof. John Perry's theory regarding the transmission of heat in boiler practice have been published, many engineers have been trying to prove to their own satisfaction that Professor Perry was in the bounds of reason when he stated that boilers should be made to do ten and possibly twenty times their usual rate.

High rate of heat transmission in boiler operation is by no means a new thing, although engineers are doubtless learning more every day about the laws and conditions governing transmission of heat.

The old two-flue boiler, which is now practically discarded, could deliver a boiler horsepower on one and one-half to two square feet of water heating surface without the slightest difficulty; in fact, the available capacity of this type of boiler was such an uncertain quantity that the only satisfactory method to pursue in replacing them with other types was to measure the actual evaporation obtained and use this as a guide in selecting the proper capacity for the new boilers. Comparison on a heating surface basis between the two-flue boiler and other types was likely to lead to most embarrassing results for the engineer responsible for such changes. The high evaporative performance of the two-flue boiler was, however, not obtained by uniformly increasing the evaporation over all the heating surface, but by extending the area of direct heating surface so as to include all of the heating surface of the boiler, for the combustion of the fuel in this type could be completed in the flues, and it was sometimes completed in the stack, if forced enough, all of which did not contribute to high efficiency, but did tremendously increase the capacity.

To make modern types of boiler do many times the work that is ordinarily expected of them, and do it with safety, it will be found necessary to have the additional work done by the heating surface not directly exposed to the fire, just as in the case of the two-flue boiler. Anyone conversant with customary boiler operation of today will not believe that the direct-heating surface of a boiler, if made of the usual materials, can possibly transmit ten times the heat that it is usually transmitting at present and still retain its physical characteristics.

Boiler-insurance companies have frowned on the various attachments for furnaces and boilers that have been brought out from time to time with promises to double capacities, and experience has taught that many such devices are detrimental to safety and seriously reduce the life of boilers on which they are used. With the agitation for, and the actual enactment of laws governing, the safe construction and operation of steam boilers, there will come a time when the maximum capacity that is allowed to be taken from a given amount of heating surface will be limited just as the steam pressure allowed is limited today. Overworking a boiler as regards the quantity of steam generated is just as dangerous as overpressure, and possibly more so, because of the insiduous manner in which defects rendering a boiler unsafe are produced by this means. Careful bookkeeping will probably show that with the possible exception of plants having a high peak load lasting for comparatively short periods each day, it is not economy to force boilers much beyond the usual rates that are obtained now. There are other items beside interest on the investment and operating charges to be considered, and reasonable safety of operation is one of these that must be given due weight.

Paying for the Right to Work

When it is demonstrated by proper examination that an engineer is competent to take care of a steam plant, whose interests are directly safeguarded, the engineer's or those of the public? Is it the purpose of a license law to prevent incompent engineers from hurting or killing themselves or to protect the public from the results of their ignorance? There is, of course, only one possible answer. Public safety is the object of all engineers' license laws, in the broad sense; protection of the engineer himself is not one of the considerations on which the laws are based. Why, then, should an

engineer be made to pay any fee for examination? Is the foreman of a powder mill required to pay a fee to any municipal authority before he is allowed to earn his living in his hazardous vocation? Not to our knowledge. Nor is any other skilled worker, no matter how much opportunity he may have for inflicting damage to life, excepting prescription clerks in pharmacies. We believe that they have to pay for their examination and pharmaceutical licenses, which is utterly unfair, but we do not know of any other classes of employee that are mulcted in this way.

The so-called argument that merchants, contractors, etc., have to pay for license to "earn a living" is sometimes dragged out as a reason why engineers should do so. Leaving aside the question of whether it is just to charge a business man for a license, the case is so obviously different from that of a man who works for a stipulated rate of pay that the "argument" is not worthy of discussion.

The bare, fundamental facts are that an engineer is examined in order to determine whether or not he will be a menace to the community if he should be allowed to control a steam plant, and a license is issued to him to show that he has given evidence of trustworthiness. The whole procedure is for the protection of the community, and the community should pay for it, just as it pays for police protection and a fire department.

The Biggest Factor

In the agitation that is going on at the present time for the enactment of laws prohibiting the construction of unsafe boilers and prescribing examination for the determination of the fitness of those who are to be intrusted with their care, the boiler operative should not lose sight of the fact that eternal vigilance on his part is one of the most necessary requisites to safety. It is unfortunate that an attendant's carefulness cannot be definitely determined by an examination.

The water column may be of the best design, its connections to the boiler may be made in the most approved manner and of ample size; but unless the attendant is careful to blow out these connections regularly, scale may stop them up and render accident inevitable.

A safety valve may be of the best make and of ample area, but unless it is tested regularly to determine its freedom, there is no certainty that it will respond to requirements when conditions arise that tend to produce an increase in the pressure beyond the safe limit.

The feed water used may be of excellent quality but very few sources of water supply remain absolutely constant, and unless frequent examinations of the interior surfaces of a boiler are made, there is no telling when scale or deposit may become a menace to safety.

The original supports of a boiler might be all that could be required from the standpoint of safety, but settings are subject to wear, and sometimes under forced conditions this wear is extremely rapid; only careful examination at frequent intervals renders a lug-supported boiler safe from accident due to change of position.

A small leak of steam coming from a seam may be a calking defect that can be remedied at leisure, but a careful examination to determine the exact nature of many such defects has shown that they were warnings of a cracked sheet, which, if neglected, would certainly lead to a disastrous explosion.

No matter how high a percentage in examination was given, what class of license was obtained, what kind of previous record a boiler attendant may have back of him or how complete or safe an equipment he may be operating, the fact still remains that his vigilance, or the lack of it, on the job is the biggest factor rendering boiler operation safe or otherwise.

Small Leaks

A steam consumption of sixteen pounds per brake horsepower-hour and an actual evaporation of eight pounds of water per pound of coal are considered a fair performance for the average plant. With coal having a heat value of 14,000 heat units per pound this would represent only about 9 per cent. of the latent energy of the coal available as mechanical energy at the flywheel of the engine—a very small proportion when one considers the numerous devices which have been introduced to improve the economy of the steam plant. The remaining 91 per cent. of the heat is lost in various ways; some passes up the stack with the waste gases; a large part is rejected in the exhaust from the engine, part of which is regained in raising the temperature of the feed water, but most of which is wasted by being given up to the condensing water, and a moderate amount of the heat is lost through radiation. In addition, there is the friction loss in the engine. These losses, however, cannot be eliminated. The best that can be done is to reduce them to a minimum by means of properly designed equipment and intelligent operation.

There are, however, other sources of waste which are entirely within the power of the engineer to avoid. These are the small leaks through blowoff valves, valves in the steam lines not in use, and the returns from neglected traps. These leaks, although apparently unimportant in themselves, nevertheless amount in the aggregate to a considerable sum in dollars and cents during a year; and in view of the very small thermal efficiency which is possible under the best conditions, any saving, however slight, is significant. A well designed plant does not insure economy; it is only a means toward that end, for after all the problem of economical operation is up to the engineer in charge.

Another Perpetual Motion Scheme Hunting for Capital

It does seem that there is no limit to human stupidity and credulity. In view of the many demonstrations which have been made of the futility of trying to get energy for nothing, one would suppose that the educated public would have better sense than to put money into that sort of a scheme. But every perpetual-motion "crank" whose chimera differs in any respect from those which have been exposed before seems to be able to hypnotize otherwise sensible persons into a condition of childlike credulity that would be highly amusing if it were not so serious a calamity.

The result of the latest effort to get something for nothing is labeled the "hydrodynamo." It is an aggregation of hydrostatic cylinders, chains, walking-beams and floats that would do credit to the genius of a mechanically-inclined lunatic but embodies a simple physical fallacy so obvious to a trained observer that we can only gasp with astonishment upon finding the thing indorsed by two men who are graduates of technical schools and have had several years of practical experience in engineering work. The "hydrodynamo," we are informed in a pamphlet issued by the Indiana Power Company, which appears to be the proud possessor of this ridiculous contraption, "runs anywhere without fuel," and the only loss is that of the friction of the parts, which amounts to "33½ per cent. of the gross effort" exerted by the machine, leaving "about 66.1 per cent. net efficient energy in the particular machine now being constructed." The slight arithmetical discrepancy in the promoter's figures is unimportant. Moreover, you could scarcely expect anyone absorbed in problems of "barodynic energy," "cognate forms," "paired antiforces," "transaxial vibrations" and such to descend to the plebeian level of a sum in subtraction; that sum was evidently "done" by the office boy.

The fallacy in everyday physics which the machine embodies will absolutely prevent it from working at all under the influence of any inherent force; it will never turn over, unless driven by some external source of power. We shall describe it in detail and point out the fallacy in an early issue. Meanwhile, we advise everyone who is invited to buy stock in the company to demand a demonstration of an actual machine under the unrestricted supervision of recognized experts before parting with a dollar or agreeing to part with one.

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it.

Air Necessary to Operate Brake

A tank of 250 cubic feet capacity is charged with compressed air at 250 pounds gage pressure. How many times will this air operate two 8x10-inch brake cylinders taking the air at a pressure of 40 pounds?

L. J. K.

Assuming that a temperature of 60 degrees Fahrenheit is maintained and that there are no losses from clearance in brake cylinders and piping at 250 pounds pressure, the tank will contain

$$250 \times \frac{265}{14.7}$$

cubic feet of free air. At 40 pounds pressure the tank will have

$$250 \times \frac{55}{14.7}$$

cubic feet of free air. The quantity of air available for operating the brake will be the difference between

$$250 \times \frac{265}{14.7} \text{ and } 250 \times \frac{55}{14.7}$$

or 3570 cubic feet of free air. The volume of the two 8x10-inch cylinders is 0.582 cubic foot and at each application of the brake would use at 40 pounds pressure

$$0.582 \times \frac{55}{14.7} = 2.18$$

cubic feet of free air, and the brake can be operated as many times as 2.18 is contained in 3570, or 1640 times.

Vacuum Required to Lift Water

I have a triplex power pump 5x8 inches running 45 revolutions per minute supposed to deliver 93 gallons per minute, but it does not throw half of this amount. I put a new vacuum gage on the suction line and it registered 19 inches. Is this enough vacuum to lift the water 27 feet?

T. S.

Your figures and vacuum gage are wrong. If the pump filled the cylinders running at the speed given 93 gallons of water would not be delivered.

Water would rise to a hight of only 21½ feet in a 19-inch vacuum and it would take 23.8 inches to raise water 27 feet. It is probable that the pump cylinders fill to less than half their capacity.

Turning the Eccentric

If I should increase the angular advance of the eccentric what effect would it have on the cutoff, and would it have any effect on the piston?

V. K. S.

Increasing the angular advance of the eccentric simply means turning it farther ahead of the crank. If this is done all of the events in the valve cycle will occur earlier in the stroke than they did before.

Changing Speed of Engine

An engine is making 75 revolutions per minute. The pulley on the engine shaft is 12 inches diameter and the governor pulley is 8 inches diameter.

What changes are necessary to increase the speed to 85 revolutions per minute?

T. E. M.

To increase the number of revolutions per minute from 75 to 85 it will be necessary to decrease the size of the pulley on the engine shaft or increase the size of the one on the governor proportionally to the desired change in speed. The proper size of pulley for the engine shaft may be found by the proportion,

85:75::12:10.58

If the governor pulley is changed the proportion will be

75:85::8:9.06.

Exhaust Steam Jacket

As the exhaust from the cylinder is warmer than the air, would it not be economical to jacket a steam-engine cylinder with it?

E. R. T.

It would be the worst practice possible as the temperature of the exhaust is not so much below that of the air in the immediate vicinity of the cylinder and as its circulation will be so rapid it will carry away a great deal more heat than the air could. A dead air space protected by nonconducting covering makes the most economical cylinder jacket.

Work Done in One Stroke of Piston

The area of an indicator diagram is 3.47 square inches. Ratio of length of diagram to length of stroke is ⅓ and the scale of spring is 40; the diameter of cylinder is 16 inches. What is the work of the engine for one stroke?

W. J. K.

The area of an indicator diagram divided by its length gives the average hight, which multiplied by the scale of the spring will give the mean effective pressure per square inch of piston area. Assuming that the length of the diagram is 4 inches the mean effective pressure acting on a 16-inch piston will be

$$\frac{3.47 \times 40 \times 201}{4} = 6974.7$$

pounds. If the stroke of the piston is six times the length of the diagram it will be 2 feet and the work done in one stroke of the engine will be

$$2 \times 6974.7 = 13{,}949.4$$

foot-pounds.

Cooling Boilers

How would you cool off boilers, and why?

H. J. M.

The damper, fire- and ashpit doors should be closed and the boiler and its contents allowed to cool slowly without the admission of cold air to any part of the setting. If the water is let out of a hot boiler and cold air comes in contact with the shell and tubes while the brickwork is hot, there will be contraction stresses which may cause leakage and perhaps serious strains.

Washing Boilers

How often should boilers be washed out?

D. A. W.

Only experience will determine for each case. In the case of a plant which is new to the operator the boiler should be opened at the first opportunity, cleaned, and its condition noted. At what in his judgment is a reasonable time thereafter the boiler should again be opened, its condition compared with the former one and used as a basis for determining the frequency of future washings.

Effect of Keying

I have keyed the rod on my engine until the wedge has reached the top of the slot and can go no farther. Should I put shims behind the stationary or the moving half of the box, and will keying alter the length of this type of rod?

S. L. G.

Shims should in this case be next to the stationary half of the brass. If the wear is equal at both ends the rod length will not change as it will lengthen at one end as much as it shortens at the other.

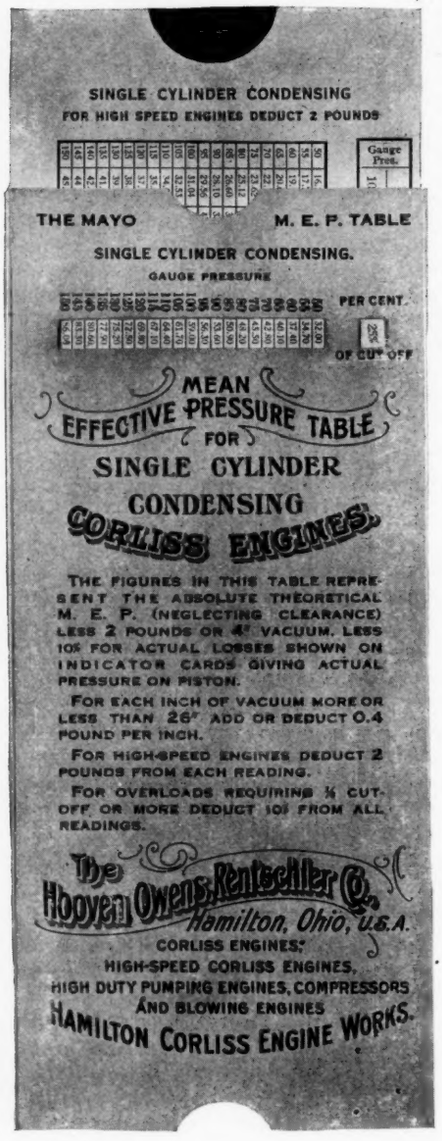
As the box at one end wears and the lost motion is taken up by moving the wedge, the pin is carried toward the end of the rod and, by increasing the distance between the pins, lengthening the rod. As the box at the other end wears, the pin is carried toward the middle of the rod by keying and the rod is shortened.

New Power House Equipment

What the inventor and the manufacturer are doing to save time and money in the engine room and power house. Engine room news.

Mayo's M. E. P. Tables

It frequently happens that the engineer and others interested in the generation and transmission of power by steam desire to know quickly and without resort to cumbersome arithmetical methods what power may be expected from an engine of given dimensions operating under certain conditions of steam pressure, point of cutoff, back pressure, vacuum, etc., or the conditions under which the engine must operate and the load to be carried may be known and the selection of an engine of proper dimensions becomes the problem to be solved.


SINGLE CYLINDER CONDENSING
FOR HIGH SPEED ENGINES DEDUCT 2 POUNDS
THE MAYO M. E. P. TABLE
SINGLE CYLINDER CONDENSING.
GAUGE PRESSURE
PER CENT.
OF CUT OFF
MEAN EFFECTIVE PRESSURE TABLE FOR SINGLE CYLINDER CONDENSING CORLISS ENGINES
THE FIGURES IN THIS TABLE REPRESENT THE ABSOLUTE THEORETICAL M. E. P. (NEGLECTING CLEARANCE) LESS 2 POUNDS OR 4" VACUUM. LESS 10% FOR ACTUAL LOSSES SHOWN ON INDICATOR CARDS GIVING ACTUAL PRESSURE ON PISTON.
FOR EACH INCH OF VACUUM MORE OR LESS THAN 26" ADD OR DEDUCT 0.4 POUND PER INCH.
FOR HIGH-SPEED ENGINES DEDUCT 2 POUNDS FROM EACH READING.
FOR OVERLOADS REQUIRING ¼ CUT-OFF OR MORE DEDUCT 10% FROM ALL READINGS.
The Hooven, Owens, Rentschler Co.
Hamilton, Ohio, U.S.A.
CORLISS ENGINES.
HIGH-SPEED CORLISS ENGINES.
HIGH DUTY PUMPING ENGINES, COMPRESSORS AND BLOWING ENGINES
HAMILTON CORLISS ENGINE WORKS.

There are countless times that a short, exact and ready-at-hand method of making power calculations would be used and appreciated if available. Making mathematical computations is a tedious and purely mechanical process, from which most men naturally shrink, and when horsepower, capacity and related data are wanted, the data log of some engine builder is hunted through for the desired information.

Search through the pages of catalogs and engineers' reference books and stub-pencil arithmetic on the back of an envelop are very neatly "side stepped" by the use of the Mayo mean effective pressure table, lately brought out by Hooven, Owens, Rentschler Company, Hamilton, O.

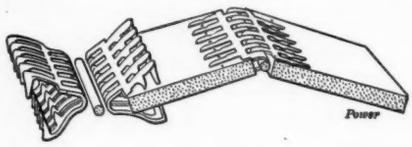
It consists of an oblong celluloid sheath with a transverse slot near the end, through which the mean effective pressure for any point of cutoff may be read from the sliding card inside the sheath.

Both sides of the sheath and the card are adapted for use, one for condensing and the other for noncondensing engines.

In the illustration the slide is set for a cutoff at 25 per cent. of the stroke and the mean effective pressure for any gage pressure from 50 to 150 pounds is instantly read through the slot.

Alligator Steel Belt Lacing

This belt fastener is adapted for use on leather, rubber, cotton or canvas stitched belting and consists of a separable lacing hinging on a rawhide or sectional steel rocker pin. The flexible joint insures quick contact with the pulley whether the belt is run with either or both sides in service, and as the lacing engages both sides of the belt alike the strain is thrown in direct line with the pull, making a joint which is easy on the texture of the belt.



ALLIGATOR STEEL BELT LACING

With rubber or textile belting this lace overcomes the tendency to fray at the ends as the members of the lace clamp the stubs so as to protect them. No punched holes are required, the teeth on each side of the lacing entering the belt on different lines of perforation; each prong has two teeth, the longer clinching through the belt and the shorter point on the opposite side giving additional strength. Both prongs enter the belt so that the pull is against the edge of the steel, thus avoiding weakening the belt. In rubber and textile belts the prongs do not cut the long threads in which lie the strength of that class of belting.

This belt lacing is manufactured by the Flexible Steel Lacing Company, 542 West Jackson boulevard, Chicago, Ill.

The Reiter Boiler Cleaner

This device, illustrated in Figs. 1 and 2, is intended to remove impurities from marine, tubular, or water-tube boilers, discharge them into a precipitator or settling chamber and return the purified water to the boiler. An upflow pipe is tapped into the boiler near the water level, where the ebullition is most rapid, and is connected with the top of a precipitating tank which provides a place for the accumulation of sediment, the purified water returning from the middle of the tank into the lower and cooler part of the boiler.

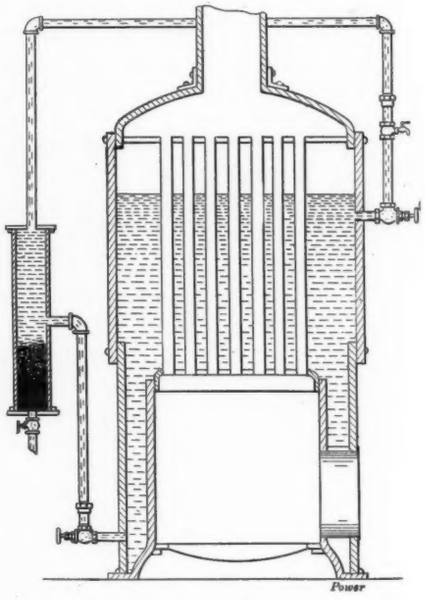


FIG. 1. CLEANER APPLIED TO A VERTICAL BOILER

There are no moving parts to the system and all valves are either gate or angle valves, which interpose the least

resistance to the circulation of water. An air cock suitably located is placed in the system at a convenient point above the water line to relieve the pipes of air when starting.

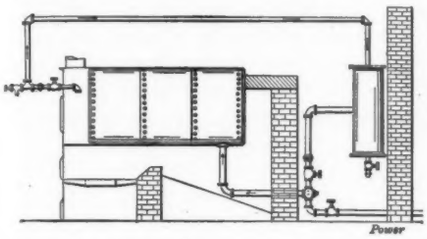


FIG. 2. CLEANER APPLIED TO A HORIZONTAL BOILER

It is claimed that this device will efficiently remove floating impurities, prevent the formation of scale and materially increase the general efficiency of a boiler installation. It is manufactured by the Reiter Boiler Cleaner Company, Elgin, Ill.

The Wattles Stoking Register

This device automatically produces a complete record, on a sensitized-paper chart, of the density of the smoke passing through a stack for every moment of each 24 hours' run.

The device serves three purposes: It records the time of stoking the furnaces, the amount of smoke passing to the stack and is responsible for a reduction in the amount of fuel consumed, because, knowing that a record is automatically

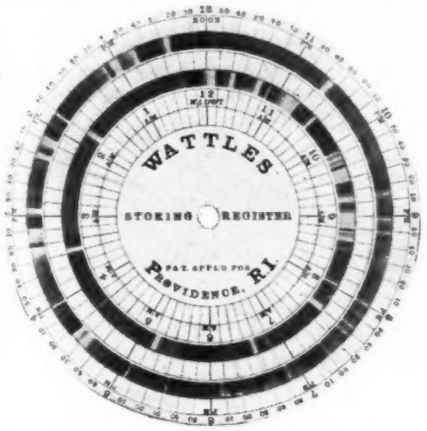


FIG. 1. CHART SHOWING FAULTY FIRING

kept of the smoke passing up the stack, the firemen are compelled to fire in a consistent, scientific manner.

The recorder consists of a box in which is placed an eight-day clock and a disk that fits on the shaft of the clock. On this disk a white sensitized-paper dial is placed, the dial being graduated into hours and minutes on two circles, as shown in Figs. 1 and 2.

The outer circle registers from 12 noon to 12 midnight; the inner circle from 12 midnight to 12 noon. The paper dial is put on the disk in a dark room and then placed in a tight case which admits no light. This case is fitted with a shutter which closes over a slot in the dial case, the slot being made the width of the recording circle. When the dial has recorded on the outer circle for 12 hours, the shutter is tripped automatically and the remaining 12 hours of the 24 are recorded on the inner circle through a second slot in the case. The device is placed and operated as follows:

On one side of the uptake, or chimney, as the case may be, is an incandescent lamp, so placed that its light will shine through a hole provided for that purpose. On the opposite side of the stack, or uptake, is another hole exactly in line with the first. The dial mechanism is placed so that the opening in the chart case will come in front of the hole. The light rays of the lamp will enter the slot in the disk case, and the sensitized paper, being exposed to the light, will turn dark when developed in the proper solution.

If nothing prevents the light rays from reaching the paper, a dark ring will show on the chart when developed, but if smoke passes between the lamp and the chart, the paper is not exposed fully to the light rays, and, consequently, it will not turn dark upon development.

The lamp is placed in a sheet-iron cone at the end away from the flue, so that it does not come in contact with the smoke, soot and hot gases. The lamp is kept clean by means of air holes placed around it, which admit enough air to fulfil this purpose.

The recording machine is placed with the lens in a second cast-iron cone, and is kept clean by air entering the cone, as in the case of the lamp.

Referring to Fig. 1, which indicates very faulty firing, it may be seen that the stoking was not only irregular during the entire 24 hours' run, but that an excessive amount of smoke passed through the uptake, not only at frequent intervals, but for long periods. The dark sections of the inner and outer rings on the chart indicate a clean stack, while the light sections indicate excessive smoke.

Fig. 2 shows a contrast, and represents careful and systematic firing. But two smoky periods show during the 24 hours, one at 10.40 a.m. and one at 11.45 a.m.

The very dark mark on the outer circle of the chart is due to the back lash of the clock gearing in taking up after changing the dial, as the paper is exposed for a longer period than when the dial has begun to turn.

This device is also a check on the voltage of the lamp, for if the voltage is allowed to rise above normal, the exposed portion of the chart will be darker than the other section. If the voltage is low, the exposed section of the chart will be lighter during that period.

In order to obtain a percentage of the smoke periods, the sum of the white spaces, in minutes, is obtained and divided by the total time the chart has recorded. If a chart is satisfactory as to smoke percentage, and shows that the

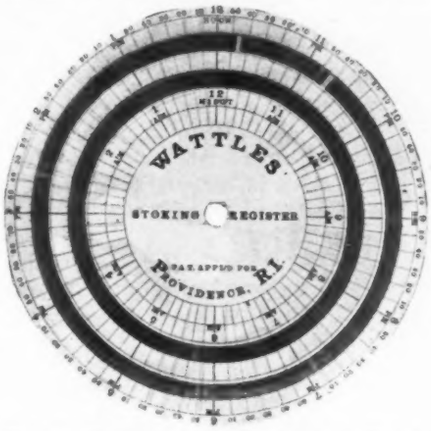


FIG. 2. CHART SHOWING CAREFUL FIRING

firemen are doing good work, it is filed; but if faulty, explanations are in order. With this chart recording the work of the firemen, a considerable saving in coal can be made, besides eliminating the smoke nuisance, as is shown by the report on coal consumption and electric energy for 1908-1909, in the Lauderdale building, Providence, R. I.

This plant has an equipment of three Babcock & Wilcox boilers of 125 horsepower capacity each, one 125-horsepower engine driving one 100-kilowatt generator, two 85-horsepower engines driving two 40-kilowatt generators.

It is operated continuously and also furnishes steam for the operation of seven hydraulic elevators, of which five are for passengers and two for freight service. The average cost of coal for the years 1908 and 1909 was $3.32 per ton.

		Amount Saved.
Coal consumption, 1908.....	2,871.50 tons	
Coal consumption, 1909.....	2,638.18 tons	
	233.32 tons at $3.32 per ton	$ 774.62
Removal of ashes, 1908.....	6,507 cans	
Removal of ashes, 1909.....	5,509 cans	
	998 cans at $0.05 per can	49 90
Electric energy, 1908.....	281,503,882 watts	
Electric energy, 1909.....	289,715,584 watts	
	8,211,762 watts increase for 1909	
Estimated cost of producing increased electric energy for 1909:		
55 tons of coal at $3.32 per ton.....		182.60
Total saving for 1909.....		$1,007.12

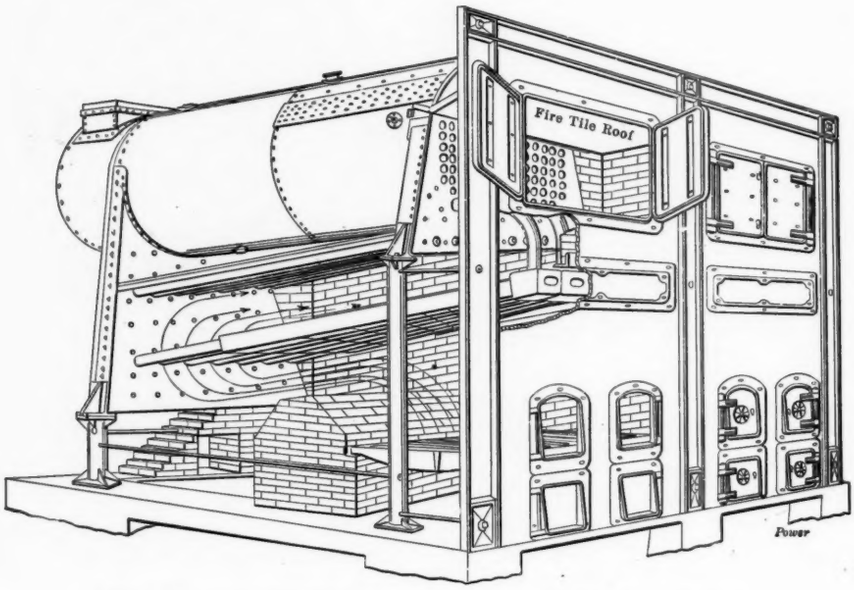
The Wattles stoker register has been in operation at this plant during 1909, saving 10½ per cent. For the first six months of 1910, it is stated that the coal consumption has remained practically unchanged, as compared with 1909. The tons of coal consumed were 1281.76, as against 1208.63 in 1909, and during this period the plants have furnished 11,000,000 additional watts.

The recorder is made by the Wattles Stoking Register Company, Butler Exchange, Providence, R. I.

The Improved Hawkes Boiler

The accompanying sketch shows in perspective the latest features of the Hawkes boiler, the essential difference from former designs being in the front head, which extends flush with the end of the boiler and forms the water leg into which the upper bank of tubes is expanded. This difference in design increases the circulating area and facilitates the steaming action. The lower bank of tubes is covered, as before, with a substantial interlocking fire-clay roofing, forming a reverberatory arch over which the hot gases must pass. A distinctive feature of the design is the independent connection of each lower tube with the upper shell, which allows expansion and contraction to take place in each tube independently.

One of the chief advantages attained by the new construction of the front water leg lies in the substantial support of the front end of the boiler which rests on cast-iron columns entirely independent of the brickwork. The shell itself does not come in contact with the fire and no joint seams or handholes are exposed to the hot furnace gases. All openings are accessible from the outside where repairs can easily be made. The furnace arrangement makes the boiler practically smokeless under any reasonable conditions of firing. The tubes may be blown in the direction of the draft. This boiler is now built by the Hawkes Boiler Company, Marquette building, Chicago, Ill.



SHOWING IMPROVEMENTS IN THE HAWKES WATER-TUBE BOILER

An Unloading Car Chute

A device for loading wagons from coal cars is illustrated herewith. It is made by the Quick Unloading Car Chute Company, Birmingham, Ala.

The device is simple and its construction will be at once comprehended from the illustrations. Fig. 1 shows a wagon loaded, ready to be hauled away. The advantages that come from this chute are that it saves the team from waiting while the shovelers in the car do their work and it enables them to work continuously instead of waiting for the arrival of a team. There being no waiting for the wagon, or wagons waiting for loads, one-third more work can be done with a given number of teams.

Fig. 2 gives an idea of the construction of the chute. It is made of sheet-steel angle iron and is hung to the top of the car by two large hooks, which may be attached to the bottom frame of the chute at three different points, according to the size of the wagon to be loaded. The chutes are adjusted by means of a chain, by the hooking up of which the front of the chute is raised, while by letting it down the chute is given a greater inclination.

The chute can be used on box cars as well as on gondolas, and can easily be handled by two men. By using two chutes in a car and dividing the distance, the shoveler can unload the car without moving the chutes.

By using this device a wagon will be detained only one minute and can then start on its return trip. If extra large wagons are used, holding more than one and a half cubic yards, the contents of two chutes can be placed in each wagon, as was done in the case illustrated in Fig. 1. The driver of the team adjusts the bolt which locks the door and discharges the load from the chute into the wagon, then leaves the door relocked and ready for the chute to be reloaded. When cars are regularly unloaded from private tracks, the chutes can be permanently mounted on supports at the side and the car placed in a position for unloading. In plants where the coal must be shoveled from the car, the chute should prove to be a great time saver.

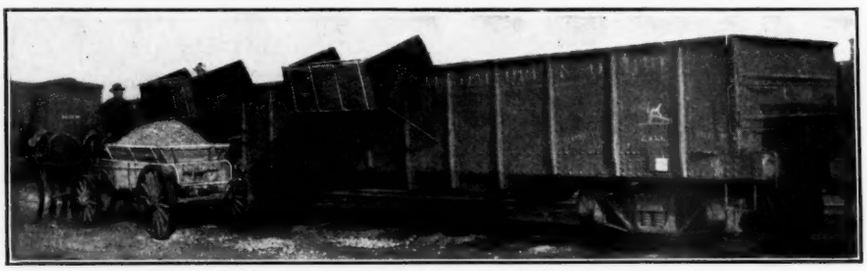
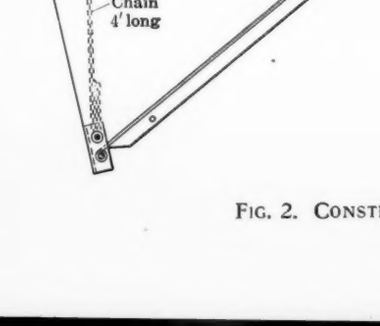


FIG. 1. WAGON LOADED IN ONE MINUTE



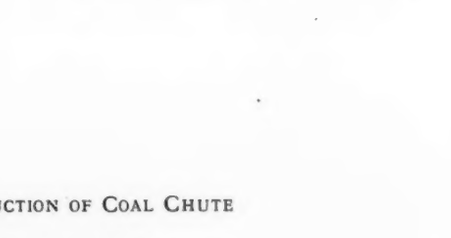


FIG. 2. CONSTRUCTION OF COAL CHUTE

PERSONAL *

C. H. Hughes & Co., mechanical and electrical engineers, formerly at 115 Broadway, New York City, have moved their quarters to larger offices at 64 Wall street.

R. W. Wells, who has been assistant for the last two years to A. E. Cluett, industrial engineer, for the Cluett, Peabody Company's factories at Troy, has been appointed master mechanic of these factories. Mr. Wells was formerly chief civilian engineer at the United States Navy Yard at Mare Island, California, under Naval Constructor Holden A. Evans.

SOCIETY NOTES

On June 18 the first steps were taken to form a permanent organization to be known as the American Society of Engineer Draftsmen, embracing every branch of the profession, including mechanical, electrical, civil, architectural, marine, sanitary, automobile and aëronautical draftsmen. The first meeting of the society was held on July 27. The formation of this organization, which was conceived by E. Farrington Chandler, designer and inventor, marks the first effort to form a national society among draftsmen.

The officers chosen are E. Farrington Chandler, president; William B. Harsel, vice-president, and Henry L. Sloan, secretary and treasurer, with headquarters at 116 Nassau street, New York City.

NEW PUBLICATIONS

How to Sample Coal and Coke is the title of Bulletin No. 4, by E. G. Bailey, Fuel Testing Company, Boston, Mass. It is issued for the purpose of assisting those interested in securing representative samples of coal and coke. The information published therein relative to this subject represents many years of experience gained in the taking of and analyzing thousands of samples. It is necessary to read the first chapter of this bulletin, which deals with the fundamental principles of sampling, before taking up the other chapters. Then, whatever succeeding chapter bears on the conditions under which samples are to be taken should be read and thoroughly mastered. These chapters deal with sampling at the mines, railroad cars, vessel cargoes, boiler plants and store piles, besides sampling coke, ashes, etc. Chapter 12 deals with the shipment of samples and tells the reader how to ship the samples to the chemist. The bulletin is sent free upon application.

*Items for this column are solicited.

Universal Craftsmen Convention

The eighth annual convention of the Universal Craftsmen Council of Engineers was held at Buffalo, N. Y., August 1 to 5, and was decidedly the largest and most successful in the history of the organization. The business meetings were held at the Iroquois hotel, the headquarters of the gathering, and a spacious parlor in this building was suitably arranged for a splendid exhibit of appliances used by engineers.

The preliminary exercises of the convention were largely attended. The meeting was called to order at eleven o'clock Tuesday morning by Joseph N. Nichols, chairman of the local committee. After an invocation by Rev. Frederick H. Coman, Mr. Nichols delivered a brief address, and then introduced John Sayles, representing Mayor Fuhrmann, who welcomed the delegates and guests to Buffalo, Grand Worthy Chief Walter S. Caldwell responding. Addresses were also made by Congressman D. S. Alexander, Robert Ingersoll and Herbert E. Stone.

At the conclusion of the addresses a memorial service was held for members who had passed away during the year. A. M. Wickens read the service, and a quartet sang "The Lord's Prayer" and "Lead, Kindly Light." The committees were then appointed and the meeting adjourned.

At the several succeeding sessions of the delegates considerable important business was transacted, and at the closing meeting the following grand officers were elected and installed: John Cope, worthy chief; Thomas H. Jones, assistant worthy chief; J. U. Bunce, secretary; John MacMorran, treasurer; Henry C. Senn, warden; P. E. Crumrine, guard; Charles F. Seigrist, ...bert Simpson, William H. Armstrong, board of trustees; John Cope, Walter S. Caldwell, John MacMorran, W. J. Ranton, Thomas H. Jones, board of governors.

There were many features of entertainment connected with the convention, including automobile rides to places of interest about the city, a visit to the Larkin Manufacturing Company, a steamboat ride to Fort Erie, a visit to Niagara Falls, a steamboat excursion on Lake Erie to Crystal beach, where a fish dinner was served, and a theater party to Shea's vaudeville house for the ladies.

The closing social event was a banquet at Statter's restaurant on Friday night. After the dinner was served, Joseph N. Nichols, the toastmaster, introduced the following gentlemen who made short addresses: Colonel Francis Ward, John Cope, Robert Ingersoll, John Moore, W. M. Williams, Joseph N. Gregory and Herbert E. Stone. Interspersed between the speeches the following entertainers appeared: Anna Batten Edwards, soprano; Frank Corbett, tenor, Consolidated Valve Company; C. F. Huie, vocal selections, Buffalo Refining Company; "Herb" Wunder, pianist, the Lunkenheimer Company; "Billy" Murray, songs, Jenkins Brothers, and "Jack" Armour, humorist, POWER.

The attendance of ladies was unusually large, and their interests were taken care of by the ladies local committee, headed by Mrs. Joseph N. Nichols.

The following firms had representatives present: The Strong-Carlisle-Hammond Company, the Purimachos Works, Oldman Boiler Works, POWER AND THE ENGINEER, Lagonda Manufacturing Company, Lake Erie Boiler Compound Company, Crane Company, Leroy Chemical Company, the Lunkenheimer Company, C. E. Squires, Buffalo Mill Supply Company, the McLeod & Henry Company, Skinner Engine Company, General Electric Company, Jones Under-Feed Stoker Company of America, *Southern Engineer,* Frontier Engine Company, John Knepper & Son, the Whittet-Barrett Company, Colza Chemical Company, Manzel Brothers, Greene, Tweed & Co., the Powell Valve Company, George W. Knowlton Rubber Company, the Buffalo Refining Company, Ashton Valve Company, Dearborn Drug and Chemical Works, Keystone Lubricating Company, Anderson Trap Company, International Steam Pump Company, Home Rubber Company, Joseph Dixon Crucible Company, Jenkins Brothers, Farrar & Trefts, McCarthy Brothers and Ford, Garlock Packing Company, Stebbins Oil Company, H. W. Johns-Manville Company, Liberty Manufacturing Company, Robertson Cataract Company, Ziermore Regulator Company, Buffalo Belting Company, Tashenburg Brothers, William A. Case & Son, Buffalo Brewers' Supply Company, Timmins & Butler, Hills-McCanna Company, Ruckel & Son, Standard Oil Company, Kempner Boiler Compound Company, Carl W. Schantz, Warren & Dellwardt Company, Howard Brothers Boiler Works, R. & P. Coal and Iron Company, Pomeroy Stelrecht, Weger Machine Company, Howard Iron Works, Hoffman Oil Company, Commercial Oil Company, Penn Oil and Supply Company, Cling Surface Company, Murphy Iron Works.

It was decided to hold the next annual meeting in Philadelphia, Penn., during week commencing Monday, August 7.

A Marvelous Accident

According to *The New York Times,* the Hudson river steamboat "Mary Powell" met with a very unusual—not to say astounding—accident the other day. The newspaper reports that "the crosshead which fastens the piston head to the slides in which it moves broke." Undoubtedly this was caused by the triple-expansion crank pin becoming fouled with the boiler eccentric and thereby straining the water-leg of the off cylinder at the base of the air pump.

NEW INVENTIONS

Printed copies of patents are furnished by the Patent Office at 5c. each. Address the Commissioner of Patents, Washington, D. C.

PRIME MOVERS

REVERSING GAS ENGINE. Daniel Schlegel, Dryville, Penn. 964,564.

ROTARY ENGINE. Emil F. Chader, Pearl River, and Vincent Anderson, Arlington, N. Y. 964,754.

POWER - GENERATING APPARATUS. Frank W. Lierman, Milwaukee, Wis.; Augusta Lierman, executrix of said Frank W. Lierman, deceased. 964,923.

ENGINE. Joseph D. Ferry, East McKeesport, Penn. 965,094.

INTERNAL COMBUSTION ENGINE. Hugo C. Gibson, New York, N. Y. 965,135.

BOILERS, FURNACES AND GAS PRODUCERS

FURNACE. Charles Sinclair Drummond, London, England. 964,502.

FURNACE. Hermann Kurtenacker, Berlin, Germany. 964,656.

BOILER SETTING. George B. McLeod, Windsor, Ontario, Canada. 964,931.

VAPOR BURNER. Emery M. Byers, Belvidere, Ill. 965,074.

GAS PRODUCER. Roscoe P. Conkling, San Luis Potosi, Mexico. 965,086.

POWER PLANT AUXILIARIES AND APPLIANCES

COOLING SYSTEM FOR INTERNAL COMBUSTION ENGINES. Thormod Odee, Beloit, Wis., assignor to Thormod Odee and A. S. Koto, copartners, doing business as Odee & Koto Manufacturing Company, Beloit, Wis. 964,542.

MEANS FOR INCREASING THE ENERGY OF STORED COMPRESSED AIR. William Horace Sodeau, Newcastle-upon-Tyne, England, assignor to Sir W. G. Armstrong Whitworth & Co., Ltd., Newcastle-upon-Tyne, England. 964,574.

HOSE COUPLING. John H. Stephens, Vernon, Tex. 964,578.

SIGHT-FEED LUBRICATOR. Floyd L. Swanberg, Cincinnati, Ohio, assignor to the D. T. Williams Valve Company, Cincinnati, Ohio, a Corporation of Ohio. 964,583.

OIL-FEEDING MECHANISM. Guido Fornaco, Turin, Italy, assignor, by mesne assignments, to F. I. A. T., Poughkeepsie, N. Y., a Corporation of New York. 964,637.

FLUE THIMBLE. Ralph Hall, Leadwood, Mo. 964,642.

STARTING MECHANISM FOR GAS ENGINES. Atwood B. Keyes, Henniker, N. H. 964,649.

VAPORIZER FOR HYDROCARBON ENGINES. Elden P. Lamb, Camden, Me., assignor to Camden Anchor-Rockland Machine Company, Rockland, Me. 964,657.

FEED-WATER CONTROLLER. Reuben M. Reynolds and Benjamin F. Page, South Bend, Ind. 964,690.

EXHAUST-VALVE OPERATING MECHANISM FOR EXPLOSIVE ENGINES. Walker H. Spayd and Fred M. Spayd, Van Wert, Ohio. 964,706.

CARBURETER. Robert E. Wynn, Brownsburg, Ind., assignor to himself, and John W. Hayden, Indianapolis, Ind., a Copartnership. 964,831.

CONDENSATION METER. Harry A. Austin, Lockport, N. Y. 964,837.

DAMPER - REGULATING MECHANISM. Benjamin C. Wickes, Auburn, N. Y. 964,892.

BOILER FEEDER. August Junger, Metz, Germany. 964,916.

HIGH-PRESSURE TURBINE PUMP. Walter L. Forward, West Berkeley, Cal., assignor to Byron Jackson Iron Works, West Berkeley, Cal., a Corporation of California. 964,963.

COMPRESSOR. John T. Love, Charlotte, N. C., assignor to William H. Flint and James W. Conway, Charlotte, N. C. 964,987.

INDICATOR FOR PRESSURE GAGES. Robert E. Fields, Fort Worth, Tex. 964,634.

COMPRESSOR AND VACUUM - PUMP VALVE. Charles Wainwright, Erie, Penn. 965,052.

PRESSURE-CONTROLLING DEVICE FOR BOILERS. John Daniel Walsh, Lockport, N. Y. 965,053.

FLUID-PRESSURE REGULATOR. William E. Beech, Ferris, Tex. 965,064.

PIPE COUPLING. James Clark, Bradford, Penn., assignor to S. R. Dresser Manufacturing Company, Bradford, Penn. 964,609.

ELECTRICAL INVENTIONS AND APPLICATIONS

SYSTEM OF MOTOR CONTROL. Ernest L. Gale, Sr., Yonkers, N. Y., assignor to Otis Elevator Company, Jersey City, N. J., a Corporation of New Jersey. 965,175.

MOTOR CONTROLLER. Herbert W. Cheney, Milwaukee, Wis., assignor to Allis-Chalmers Company, a Corporation of New Jersey. 965,280.

ELECTRIC WATER HEATER. Frank A. Robinson, Albany, N. Y., assignor to William G. Van Loon, Albany, N. Y. 965,333.

AUTOMATIC POTENTIAL REGULATOR FOR DYNAMOS. John J. Mueller, Jr., Peru, Ill. 965,396.

ELECTRIC MOTOR. Frank L. Sessions, Columbus, Ohio, assignor, by mesne assignments, to the Jeffrey Manufacturing Company, a Corporation of Ohio. 965,543.

ELECTRIC PUMP GOVERNOR. Charles W. Townsend, St. Louis, Mo., assignor to the Westinghouse Air Brake Company, Pittsburg, Penn., a Corporation of Pennsylvania. 965,609.

CONTROLLING MEANS FOR ELECTRIC MOTORS. George H. Whittingham, Baltimore, Md., assignor to Monitor Manufacturing Company, of Baltimore, Md., a Corporation of Maryland. 965,674.

MOTOR STARTER. George H. Whittingham, Baltimore, Md., assignor to Monitor Manufacturing Company, of Baltimore, Md., a Corporation of Maryland. 965,675.

ELECTRIC-LIGHT FIXTURE. Edwin E. Hayward, Waterbury, Conn., assignor to the Plume & Atwood Mfg. Co., Waterbury, Conn., a Corporation of Connecticut. 965,710.

ELECTRIC CABLE. William Gelpel, Southwark, England. 965,810.

ARC LAMP. George M. Little, Pittsburg, Penn., assignor to Westinghouse Electric and Manufacturing Company, a Corporation of Pennsylvania. 964,664.

BRACKET FOR COMBINATION ELECTRIC AND GAS FIXTURES. James D. Coder, Elizabeth, N. J. 964,493.

STARTING DEVICE FOR ELECTRIC LAMPS. George M. Little, Pittsburg, Penn., assignor to Westinghouse Electric and Manufacturing Company, a Corporation of Pennsylvania. 964,665.

ELECTROMAGNET PROPELLING SYSTEM. Rexford Dean, Findlay, Ohio. 964,498.

HOT-WIRE ELECTRICAL MEASURING INSTRUMENT. John T. Irwin, London, England. 964,521.

ELECTRIC SWITCH. William S. McLewee, Yardley, Penn. 964,536.

INSULATOR. John Russell Tufts, Jr., Weymouth, Mass., assignor of one-half to F. W. Clark, Boston, Mass. 964,586.

FUSE. Frederick H. Weston, Jr., Schenectady, N. Y., assignor to General Electric Company, a Corporation of New York. 964,592.

ALTERNATING-CURRENT MOTOR. Per Utne, New York, N. Y., assignor to the Union Switch and Signal Company, Swissvale, Penn., a Corporation of Pennsylvania. 964,714.

STORAGE-BATTERY INDICATOR. Paul Max Marko, New York, N. Y. 964,994.

STORAGE BATTERY. Paul Max Marko, New York, N. Y. 964,994.

PROCESS OF PURIFYING WATER. John E. Angell, St. Louis, Mo. 965,148.

CHECK VALVE. Emmet P. Gray, Detroit, Mich. 965,178.

BELTING. Varney K. Sturges, Detroit, Mich., assignor to Joseph H. Schulte and Harry Kaiser, Detroit, Mich. 965,250.

METALLIC ROD PACKING. Charles L. Trapp, South Omaha, and Otto C. Gehrman, Omaha, Neb., assignors of one-fourth to Frank J. Meyers, Omaha, Neb. 965,255.

POWER PLANT TOOLS

LIFTING JACK. Robert M. Fraley, Champaign, Ill. 964,905.

BALL-BEARING JACK SCREW. Oscar E. Tidd, Georgetown, Mass. 964,824.

RATCHET DRILL. Domenic Cortese, Spooner, Wis. 964,616.

INSULATION-CUTTING PLIERS. Harry A. Adams, East Hartford, Conn. 964,600.

COMBINATION TOOL. Norman McAslan, Yorkshire, Iowa. 964,532.

UNIVERSAL SQUARE. De Witt C. Meeker, Carthage, Mo., Mary Ann Meeker, administratrix of said De Witt C. Meeker, deceased. 965,312.

DIE STOCK. Herman W. Oster, Cleveland, Ohio, assignor to the Oster Manufacturing Company, Cleveland, Ohio, a Corporation of Ohio. 965,320.

DIE STOCK. Herman W. Oster, Cleveland, Ohio, assignor to the Oster Manufacturing Company, Cleveland, Ohio, a Corporation of Ohio. 965,403.

ENGINEERING SOCIETIES

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Pres., George Westinghouse; sec., Calvin W. Rice, Engineering Societies building, 29 West 39th St., New York. Monthly meetings in New York City.

NATIONAL ELECTRIC LIGHT ASSOCIATION

Pres., W. W. Freeman, Brooklyn, N. Y.; sec., T. C. Martin, 33 West Thirty-ninth St., New York.

AMERICAN SOCIETY OF NAVAL ENGINEERS

Pres., Engineer-in-Chief Hutch I. Cone, U .S. N.; sec. and treas., Lieutenant Henry C. Dinger, U. S. N., Bureau of Steam Engineering, Navy Department, Washington, D. C.

AMERICAN BOILER MANUFACTURERS' ASSOCIATION

Pres., E. D. Meier, 11 Broadway, New York; sec., J. D. Farasey, cor. 37th St. and Erie Railway, Cleveland, O.

WESTERN SOCIETY OF ENGINEERS

Pres., J. W. Alvord; sec., J. H. Warder, 1735 Monadnock Block, Chicago, Ill.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA

Pres., E. K. Morse; sec., E. K. Hiles, Oliver building, Pittsburg, Penn. Meetings 1st and 3d Tuesdays.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Pres., L. B. Stillwell; sec., Ralph W. Pope, 33 W. Thirty-ninth St., New York. Meetings monthly, excepting July and August.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pres., Prof. J. D. Hoffman; sec., William M. Mackay, P. O. Box 1818, New York City.

NATIONAL ASSOCIATION OF STATIONARY ENGINEERS

Pres., William J. Reynolds, Hoboken, N. J.; sec., F. W. Raven, 325 Dearborn street, Chicago, Ill. Next convention, Rochester, N. Y., September 12-17, 1910.

UNIVERSAL CRAFTSMEN COUNCIL OF ENGINEERS

Grand Worthy Chief, John Cope; sec., V. Bunce, Hotel Statler, Buffalo, N. Y. Next annual meeting in Philadelphia, Penn., week commencing Monday, August 7, 1911.

AMERICAN ORDER OF STEAM ENGINEERS

Supr. Chief Engr., Frederick Markoe, Philadelphia, Pa.; Supr. Cor. Engr., William S. Wetzler, 753 N. Forty-fourth St., Philadelphia, Pa. Next meeting at Philadelphia, June, 1911.

NATIONAL MARINE ENGINEERS BENEFICIAL ASSOCIATIONS.

Pres., William F. Yates, New York, N. Y.; sec., George A. Grubb, 1040 Dakin street, Chicago, Ill. Next meeting, St. Louis, Mo., January 16-21, 1911

OHIO SOCIETY OF MECHANICAL ELECTRICAL AND STEAM ENGINEERS

Pres., O. F. Rabbe; sec. and treas., Prof. F. E. Sanborn, Ohio State University, Columbus, Ohio.

INTERNATIONAL MASTER BOILER MAKERS' ASSOCIATION

Pres., A. N. Lucas; sec., Harry D. Vaught, 95 Liberty street, New York. Next meeting at Omaha, Neb., May, 1911.

INTERNATIONAL UNION OF STEAM ENGINEERS

Pres., Matt. Comerford; sec., Robert A. McKee 606 Main St., Peoria, Ill. Next convention, Denver, Colo., September, 1910.

NATIONAL DISTRICT HEATING ASSOCIATION.

Pres., G. W. Wright, Baltimore, Md.; sec. and treas., D. L. Gaskill, Greenville, O.